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President.....J. F. Bertram
Vice-President.....W. J. Atkinson
Secretary-Treasurer.....A. K. Piercy
Board of Governors: Einar Anderson, W. J. Atkinson, J. F. Bertram, C. M. Fleming, Ernest Lambert, A. K. Piercy, J. R. Stephenson

* Filled unexpired term.

Massachusetts

Organized 1912

Headquarters, Boston

President.....John Bonner
Vice-President.....G. B. Torrens
Secretary.....L. R. Geissenhainer
Treasurer.....A. L. Hare
Board of Governors: W. J. Ahearn, D. W. Blair, John Bonner, C. H. Dow, G. D. Fife, L. R. Geissenhainer, A. L. Hare, L. P. Licandro, F. A. Merrill, D. C. Miller, G. B. Torrens

Memphis

Organized 1944

Headquarters, Memphis

President.....R. E. Larkin
Vice-President.....G. C. Burr
Secretary.....A. T. Bevil
Treasurer.....H. H. Wilson
Board of Governors: C. S. Fischer, O. S. Humphrey, N. C. Ledbetter, W. E. Thorpe

Miami Valley

Organized 1950

Headquarters, Dayton, Ohio

President.....C. D. Weaver
Vice-President.....D. L. Bergman
Secretary.....R. W. Kimmel
Treasurer.....W. R. Budde
Board of Governors: R. C. Allen, T. I. Byrd, F. S. Hamer, G. L. Leupold

Michigan

Organized 1916

Headquarters, Detroit

President.....G. W. Akers
Vice-President.....C. A. Strand
Secretary.....R. H. Oberschulte
Treasurer.....E. F. Glanz
Board of Governors: G. W. Akers, G. L. Davis, Jr., C. F. Donohoe, D. S. Falk, D. L. McConachie, R. H. Oberschulte, C. A. Strand

Minnesota

Organized 1918

Headquarters, Minneapolis

President.....E. F. Snyder, Jr.
Vice-President.....William Sturm
Secretary.....J. A. Holbrook
Treasurer.....V. E. Pearson
Board of Governors: A. B. Algren, C. E. Gausman, G. J. Kendrick

Montreal

Organized 1936

Headquarters, Montreal, Que.

President.....J. G. Chenevert
Vice-President.....W. G. Hole
Secretary.....B. J. Horsburgh
Treasurer.....D. L. Lindsay
Board of Governors: J. C. Hennessy, R. J. Kerr, R. R. Noyes, R. A. Standring

Local Chapter Officers—1951 *(Continued)*

Nebraska

Organized 1940

Headquarters, Omaha

President.....K. E. Martin
Vice-President.....S. W. Black
Secretary.....O. J. Smith
Treasurer.....M. K. Rush
Board of Governors: S. W. Black, G. W. Colburn, W. B. Howard, K. E. Martin, B. G. Peterson, M. K. Rush, O. J. Smith

New York

Organized 1911

Headquarters, New York

President.....Ernst Graber
Vice-President.....P. B. Gordon
Secretary.....Carl H. Flink
Treasurer.....W. M. Heebner
Board of Governors: P. B. Gordon, J. B. Hewett, C. F. Kayan, J. E. Schechter, R. L. Stinard, Clifford Strock

North Carolina

Organized 1939

Headquarters, Durham

President.....R. O. McGary
Vice-President.....L. L. Vaughan
Secretary-Treasurer.....J. A. Rice
Board of Governors: M. F. DuChateau, Alton Skinner, Jr., W. H. Sullivan, Jr., R. M. Warren, Jr.

North Texas

Organized 1938

Headquarters, Dallas

President.....R. G. Lyford
Vice-President.....R. E. Allison
Secretary.....P. N. Vinther
Treasurer.....M. W. Brown
Board of Governors: Herman Blum, Jr., C. R. Gardner, H. G. Gregerson

Northeastern Oklahoma

Organized 1948

Headquarters, Tulsa

President.....R. F. Shoemaker
Vice-President.....J. N. Watt
Secretary.....F. M. Thomas
Treasurer.....C. H. Dollmeyer, Jr.
Board of Governors: E. R. Cooke, A. D. Holmes, V. W. Miles

Northern Ohio

Organized 1916

Headquarters, Cleveland

President.....John Richmond
Vice-President.....R. E. Sherman
Secretary.....R. G. Huebscher
Treasurer.....L. C. Burkes
Board of Governors: J. M. Black, D. E. Mannen, W. M. Rowe

Oklahoma

Organized 1935

Headquarters, Oklahoma City

President.....J. H. Carnahan
Vice-President.....S. J. White
Secretary.....W. J. Collins, Jr.
Treasurer.....F. X. Loeffler, Jr.
Board of Governors: H. S. Shafer, J. H. Spaan, Jr., R. E. Swan

Ontario

Organized 1922

Headquarters, Toronto

President.....William Philip
Vice-President.....J. H. Ross
Secretary-Treasurer.....H. R. Roth
Board of Governors: D. L. Angus, M. C. Bailey, J. H. Fox, N. W. Kingsland

Oregon

Organized 1939

Headquarters, Portland

President.....H. W. McKenzie
Vice-President.....K. H. Hanson
Secretary.....R. P. Lankow
Treasurer.....W. R. Pindell
Board of Governors: Dick Blankenship, R. C. Chewning, B. M. Thornburg

Pacific Northwest

Organized 1928

Headquarters, Seattle, Wash.

President.....R. M. Stern
1st Vice-President.....W. B. Pride
2nd Vice-President.....R. R. Kirkwood
Secretary.....H. T. Orebaugh
Treasurer.....T. C. Caskey
Board of Governors: R. C. Lewis, M. W. McKinstry

Philadelphia

Organized 1916

Headquarters, Philadelphia

President.....E. K. Wagner
1st Vice-President.....M. E. Barnard
2nd Vice-President.....C. F. Dietz
Secretary.....A. M. Robinson
Treasurer.....L. M. Church
Board of Governors: M. E. Barnard, L. M. Church, C. F. Dietz, C. J. Lubking, J. W. McElgin, A. M. Robinson, W. S. Scott, E. K. Wagner

Pittsburgh

Organized 1919

Headquarters, Pittsburgh

President.....W. D. Simpson
Vice-President.....B. B. Reilly
Secretary.....E. H. Riesmeyer, Jr.
Treasurer.....C. L. Benn
Board of Governors: H. A. Biber, A. F. Metzger, O. L. Williams

Local Chapter Officers—1951 *(Concluded)*

Rocky Mountain

Organized 1944

Headquarters, Denver, Colo.

President.....B. H. Spurlock, Jr.
Vice-President.....N. H. Brickham
Secretary.....J. W. Braak
Treasurer.....J. F. Mohan
Board of Governors: F. C. Allen, Fred Janssen, R. W. Petersen

St. Louis

Organized 1918

Headquarters, St. Louis

President.....L. L. Hamig
1st Vice-President.....C. H. Burnap
2nd Vice-President.....H. C. Sharp
Secretary.....J. F. Naylor, Jr.
Treasurer.....G. H. Bemarkt
Board of Governors: F. E. Ince, J. B. Killebrew, J. S. Rosebrough, H. F. Wilson

Shreveport

Organized 1948

Headquarters, Shreveport

President.....S. W. Beaty
Vice-President.....R. M. Hood
Secretary.....W. S. Evans
Treasurer.....M. A. Hudson
Board of Governors: S. W. Beaty, W. S. Evans, L. C. Haas, R. M. Hood, P. O. Rottmann, B. E. Segall, Jr., J. B. Struwe

South Texas

Organized 1938

Headquarters, Houston

President.....R. J. Salinger
Vice-President.....C. L. Fleming
Secretary.....E. G. Floeter, Jr.
Treasurer.....J. W. Holland
Board of Governors: H. W. Broadwell, J. C. Lewis, F. M. Neil

Southern California

Organized 1930

Headquarters, Los Angeles

President.....J. S. Earhart
Vice-President.....L. B. Davenport
Secretary.....C. D. Walz
Treasurer.....J. L. McCullough
Board of Governors: V. J. Burke, M. C. Greiner, E. D. Pierce, L. H. Wagenbrenner, D. D. Wile

Southwest Texas

Organized 1946

Headquarters, San Antonio

President.....I. W. Wilke
Vice-President.....W. E. Long
Secretary-Treasurer.....D. E. Locher
Board of Governors: F. C. Benham, J. G. Hopping, Jr., L. S. Pawkett

Utah

Organized 1944

Headquarters, Salt Lake City

President.....A. R. Curtis
Vice-President.....D. R. Wilde
Secretary-Treasurer.....M. L. Gollaher
Board of Governors: E. L. Baker, C. E. Ferguson, E. V. Gritton, E. D. Smith, W. L. Stuewe, E. J. Watts

Virginia

Organized 1946

Headquarters, Norfolk

President.....J. F. Boventon
Vice-President.....D. E. Phillips
Secretary-Treasurer.....J. E. Harding
Board of Governors: E. R. Ervin, J. E. Harding, J. F. Reynolds, W. P. Robinson

Washington, D. C.

Organized 1935

Headquarters, Washington, D. C.

President.....S. R. Allen
Vice-President.....G. R. Walz
Secretary.....P. H. Loughran, Jr.
Treasurer.....J. L. Manheim
Board of Governors: I. M. Bortman, Rutcher Skagerberg, F. M. Thuney

Western Michigan

Organized 1931

Headquarters, Grand Rapids

President.....V. H. Hill†, W. C. DeRoo*
Vice-President.....W. C. DeRoo
H. R. Limbacher*
Secretary.....H. R. Limbacher
P. D. Winchester*
Treasurer.....W. C. DeRoo
Board of Governors: V. C. Dean, K. E. Robinson, L. A. Tilford

Western New York

Organized 1919

Headquarters, Buffalo

President.....T. F. Killeen
1st Vice-President.....J. M. Quackenbush
2nd Vice-President.....C. W. Stone
Secretary.....R. E. Lang
Treasurer.....B. C. Candee
Assistant Treasurer.....C. W. Kaupp
Board of Governors: M. C. Beman, J. H. Bryce, Joseph Davis, W. R. Heath, S. M. Quackenbush

Wisconsin

Organized 1922

Headquarters, Milwaukee

President.....H. F. Brinen
Vice-President.....J. A. Lofte
Secretary.....L. C. Plaehn
Treasurer.....N. E. Hill
Board of Governors: D. W. Nelson, F. J. Nunlist, C. H. Randolph

† Moved.
* Filled unexpired term.

Student Branch Officers—1951

Louisiana Polytechnic Institute

Organized 1949

Headquarters, Ruston

President.....William McMullen
Vice-President.....Vernon Ashworth
Secretary.....R. L. Gorton
Treasurer.....B. A. Rose

North Carolina State College

Organized 1948

Headquarters, Raleigh

President.....Sydney Narvey
Vice-President.....Leonard Rubin
Secretary.....E. A. Travers
Treasurer.....W. G. Lloyd
Reporter.....L. R. Wensil, Jr.

Oklahoma A & M College

Organized 1950

Headquarters, Stillwater

President.....R. E. Metzkier, W. E. Suchan, Jr.
Vice-President.....J. J. Murov, Norman Peck
Secretary.....W. D. Jenkins
Treasurer.....R. J. Gorenflo, L. O. Crooks

Oregon State College

Organized 1949

Headquarters, Corvallis

President.....D. W. Blair
Vice-President.....R. E. Bulletset
Secretary-Treasurer.....C. D. Cannon

Purdue University

Organized 1948

Headquarters, W. Lafayette, Ind.

Chairman.....Stanley Kalfus, N. J. Kirch
Vice-Chairman.....Stanley Jacobs
R. O. Bement
Secretary,.....June Cox, H. F. Allen, Jr.
Reporter.....J. N. Kirch, Melvin Kalfus
Coordinator.....G. M. Bender
A. M. Landerman

Texas A & M

Organized 1946

Headquarters, College Station

President.....J. E. Schumann, J. O. Kadel
Vice-President.....W. E. Richard
W. E. Weatherby
Secretary.....A. G. Seelke, Jr., W. C. Haggard
Treasurer.....C. F. Cogan, Charles Luedtke

University of Detroit

Organized 1949

Headquarters, Detroit

President.....G. H. Williams
Vice-President.....T. J. Feahenv, Jr.
Secretary-Treasurer.....R. H. Drake
Reporter.....G. E. Sanctorum

University of Texas

Organized 1949

Headquarters, Austin

President.....W. A. Cox
Vice-President.....J. W. Paxton
Secretary.....M. F. Klevar
Treasurer.....L. T. Olsen

University of Toronto

Organized 1951

Headquarters, Toronto, Ont.

Chairman.....H. F. Spragg
Vice-Chairman.....M. S. Warmski
Secretary-Treasurer.....J. E. G. Blaiklock

1406

TRANSACTIONS

of

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1406

FIFTY-SEVENTH ANNUAL MEETING, 1951

PHILADELPHIA, PENNSYLVANIA

THE 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held in Philadelphia, Pa., with a total registration of 1746, making this one of the largest annual meetings ever held by the Society. The registration consisted of: members 1009, guests 433, and ladies 304. In conjunction with the meeting, the 10th International Heating and Ventilating Exposition, at Convention Hall, was visited by 20,000 engineers, manufacturers, contractors, architects and others.

Six technical sessions were held, one of which was a symposium on the subject *Man and His Relationship To Air*. The meetings and other functions were held at the Bellevue-Stratford, the Benjamin Franklin, and the Warwick Hotels.

Highlights of the meeting were the get-together luncheon on January 22nd at the Bellevue-Stratford, with Dr. Lillian M. Gilbreth speaking on the subject *Management Problems In The Home;* Monday evening entertainment *A Night With the Pennsylvania Dutch* at the Benjamin Franklin Hotel; and the Annual Banquet in the grand ballroom of the Bellevue-Stratford on January 24th which presented Cameron Ralston, Gettysburg, with an address entitled *The Plagues of Democracy*. The past president's emblem was presented to retiring president Lester T. Avery by past president and charter member Homer Addams. President Avery announced that the F. Paul Anderson Medal had been awarded to Samuel R. Lewis, Chicago, Illinois, past president and life member of the Society and would be presented at a later meeting.

FIRST SESSION, MONDAY, JANUARY 22, 9:30 A.M.

President Lester T. Avery called the 57th Annual Meeting to order at 9:30 A.M. and introduced John W. McElgin, Philadelphia Chapter president, who welcomed the members and guests.

President Avery, on behalf of the Society officers and members, thanked Mr. McElgin for his cordial welcome and noted that the last meeting in Philadelphia had been at the beginning of World War II in 1942. The war had forced cancellation of the 7th International Heating and Ventilating Exposition, but following its end several shows had been held and today the Society was about to open the 10th Exposition. President Avery then presented his official report. In his report the president described the growth of the Society during the previous year both in members and in chapters. He noted the loss of three Society past presidents and leaders, Dr. E. Vernon Hill, Chicago, Ill.; Dr. Willis H. Carrier, Syracuse, N. Y.; and William H. Driscoll, Syracuse, N. Y., and Jersey City, N. J. President Avery discussed his chapter visits and contemplated the growth possibilities for the Society during the coming 5 years. He stated that through the chapter meetings the Society had a great opportunity to serve the public, and urged chapter members to bring in other professional and business groups when chapter speakers had a subject of particular interest. He concluded by saying that the papers and discussion at the meeting, together with the exhibits at the International Heating and Ventilating Exposition would all help to further the aims of the Society.

REPORT OF COUNCIL

As required by Section 46 of the Membership Corporations' Law of the State of New York, the following report is presented and filed herewith.

Assets

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, etc. at New York and Cleveland	38,553.40
On Deposit in New York and Cleveland Banks	131,593.89
Securities (United States and Canadian Government Bonds) with Bankers Trust Company, New York, as Custodian	100,713.05
Accounts Receivable	18,174.46
Inventory Items	14,679.96
Deferred Items	4,668.12
Miscellaneous	2,844.23
TOTALS	$392,240.00
1949 TOTAL	362,468.28
INCREASE	$ 29,771.72

The sources of Society Cash Receipts were: Admission Fees and Dues $113,169.57, Publications $155,787.55, from Research $143,528.20, Interest $1,049.89 and Miscellaneous $290.00, making a total of $413,825.21. Cash Disbursements for Meetings, Committees and Chapters were $25,703.30, Publications $112,253.45, Headquarters $94,287.00, Research $174,636.17—Total $406,879.92.

The Society's Reserve Fund was increased by the addition of $6,345.50 Admission Fees and $10,000.00 from the General Fund and Interest Earned on Savings Accounts and Securities of $1,431.48 in the Reserve Fund. Investment of $13,486.50 of the Reserve Fund in U. S. Government Savings Bonds Series "F" was made.

Now in the Custodian Account of the Bankers Trust Company, New York, there are Securities of the United States and Canadian Governments—General Fund which cost $36,110.75, Reserve Fund $53,631.50 and F. Paul Anderson Fund $1,000.00. The Research Reserve Fund was reduced during the year by $10,000.00 to $21,252.46 to meet operating needs.

Liabilities

The Society has the following liabilities:

Federal Withholding Taxes	$ 552.70
Accrued Salaries and Advertising Commissions	11,597.66
Deferred Research Projects	9,156.89
Prepaid Admission Fees and Prepaid Dues	14,585.48
Reserve for TRANSACTIONS	28,646.94
Net Worth	327,700.33
TOTAL	$392,240.00

At the end of the fiscal year, October 31, 1950, the Balance Sheet shows the Society's Net Worth as $327,700.33 compared to $300,797.28 on October 31, 1949, a net gain of $26,903.05.

The Property funds covering the Laboratory Land and Buildings at Cleveland, Laboratory Equipment and Tools, Furniture and Fixtures at Headquarters, New York and Cleveland is $119,566.29.

Fire Insurance of $151,000.00 is carried on Buildings, $40,000.00 on Contents, Sprinkler $11,000.00, also Comprehensive Bodily Injury $100,000.00-$300,000.00 and Auto Property Damage $5,000.00. Fidelity Bonds on Officers and all employees $20,000.00. Workmen's Compensation is carried in New York and Ohio as required by law.

The Society has contributed to the Employees Retirement Plan Trust $2,357.55 and the participating employees contributed a similar amount.

Total Cash in New York and Cleveland Banks $131,593.89, consisting of $84,762.91 in the general operating fund, $6,454.23 in the Reserve Fund, $138.47 in the F. Paul Anderson Fund, $18,311.72 in the Research Fund, $21,252.46 in Research Reserve, and $674.10 in Research Endowment.

Membership

The Society admitted the following members in the grades indicated since January 1, 1950:

Members	216
Associate Members	94
Affiliates	190
Junior Members	305
Students	399
TOTAL	1,204

The names and addresses of the candidates for membership were published in the Journal of the Society each month during the year and are on record in the Secretary's office. The present membership total is 8,312.

LESTER T. AVERY, *President*
REG F. TAYLOR, *Treasurer*

Dated New York, N. Y., January 22, 1951

After President Avery's report, Alfred E. Stacey, Jr., presented a special resolution on the passing of Dr. E. Vernon Hill, Mr. William H. Driscoll, and Dr. Willis H. Carrier, past presidents of the Society who had died during the year 1950.

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has learned with deep sorrow of the passing on May 15, 1950 of Dr. E. Vernon Hill, their beloved fellow member and past president, and

WHEREAS, the Society has suffered the loss of a member of great distinction, one of the first of the medical fraternity to associate himself with the Society in the basic physiological approach for human comfort, a brilliant scientist whose accomplishments have received wide recognition, and

WHEREAS, in his passing, the Society has suffered the loss of one of its outstanding members who was greatly respected for his pioneering in the medical approach to problems of the Society and his continuing activity and energy in connection therewith, and

WHEREAS, he will be remembered for his guidance and wise counsel in the early years of the Society's research, his kindliness, his active mind and his wide interest, and

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has learned with deep sorrow of the death on September 6, 1950 of their beloved member, William H. Driscoll, whose passing has taken one of the veterans in Society service, whose span of accomplishments covered nearly a half century, and who served as president and in manifold capacities through the years, and

WHEREAS, the Society has lost a man possessed of a deep sense of service as well as great engineering and executive ability, and

WHEREAS, in his passing the Society has suffered the loss of his unselfish devotion to its work, his vision, and tremendous energy and enthusiasm which have left a lasting impression on the development and welfare of the Society, and

WHEREAS, his fellow members will miss his genial presence and his generous advice and wise counsel, and

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has learned with sincere sorrow of the passing on October 7, 1950 of Dr. Willis H. Carrier, their beloved fellow member and past president, and

WHEREAS, the Society has suffered the loss of a member of great distinction; a pioneer in air conditioning; a brilliant engineer and scientist whose accomplishments have received world wide acclaim, and

WHEREAS, in his passing the Society has lost one of its outstanding members who was greatly admired as a man and revered as a master in the field he had chosen for his life work, and

WHEREAS, he will be remembered for his modesty, kindliness, and loyalty as a friend, whose many splendid traits of character endeared him to all of our members,

THEREFORE, BE IT RESOLVED THAT WE, the members of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, assembled at the 57th Annual Meeting at Philadelphia, Pa., hereby express our profound sorrow, our sense of great loss, and the high esteem in which their memory is held, and

Be It Further Resolved That this resolution of our esteem and heartfelt sympathy be placed in the Proceedings of the Society and a copy be sent to their families.

Respectfully Submitted,

S. R. Lewis, *Chairman*
Homer Addams
A. E. Stacey, Jr.

The resolutions were adopted by a rising vote during which an interval of silence was maintained in memory of these distinguished Society members.

Reg F. Taylor presented the report of the treasurer in which he compared the budget and finances of the Society in 1914 to the budget and expenditures of the fiscal year 1951. He also noted the growth in membership and the increase in net worth of the Society, and felt that the Society could be proud of its accomplishments. The Society members could take pride in the service that they had rendered their fellow-man, and could look forward to continuing and increasing that service.

Accountant's Report

FRANK G. TUSA & CO.
Certified Public Accountants
37 Wall St.
New York 5, N. Y.

The American Society of Heating and Ventilating Engineers
51 Madison Avenue
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of The American Society of Heating and Ventilating Engineers—New York and Cleveland and its related funds for the fiscal year ended October 31, 1950, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1950. For the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements, and the disbursements were supported by payment vouchers. The dues from members and the interest from investments were accounted for.

A balance sheet reflecting the financial condition of the Society as of the close of business October 31, 1950 is submitted herewith and your attention is directed to the following comments thereon:

Cash

The cash on deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the cash on hand was counted.

Marketable Securities

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safekeeping. Securities have been included in the accompanying balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1950 were classified and aged as follows:

DUES RECEIVABLE

MEMBERSHIP	Amount
Members	$ 4,204.50
Associates	8,986.76
Affiliates	431.25
Juniors	1,646.50
Students	540.00
TOTAL	$15,809.01

AGING	
1950	$15,285.51
1949	451.50
1948	61.00
1947	11.00
TOTAL	$15,809.01

SUNDRY DEBTORS

Unpaid charges made during October, 1950	$2,153.69
Unpaid charges made during September, 1950	945.47
Unpaid charges made prior to September, 1950	3,194.97
TOTAL	$6,294.13

After writing off all receivables known to be uncollectible, it is our opinion that the reserves for dues and accounts receivable doubtful of collection reflected in the accompanying balance sheet are ample to cover collection losses that may be incurred.

RESEARCH GENERAL FUND ACCOUNT RECEIVABLE

The balance due to the Research General Fund from the Society as of October 31, 1950 represents 40 percent of Dues Receivable from Members, Associates and Affiliates as of such date.

INVENTORIES

The following inventories were verified either by count or direct communciation:

TRANSACTIONS	$ 3,649.08
Emblems	299.75
GUIDE Paper	10,731.13
TOTAL	$14,679.96

A Schedule of TRANSACTIONS inventoried follows:

Volume	Year	Quantity	Price	Amount
1-48	1895-1942	3,112	$ .40	$1,244.80
49	1943	224	1.39	311.36
50	1944	177	1.82	322.14
51	1945	332	1.91	634.12
52	1946	404	1.67	674.78
53	1947	261	1.77	461.98
TOTAL				$3,649.18

Deposit Receivable

The deposit placed with the United Air Lines in the sum of $425.00 was verified by direct communication.

Advances

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of $1,433.46 and $985.77 respectively. This indebtedness is to be reimbursed by the trust during the ensuing fiscal year.

Deferred Charges

The following prepaid expenses comprise the deferred charges included as assets in the accompanying balance sheet:

Prepaid H.P.A.C. Subscriptions	$2,550.61
Prepaid Advertising Sales Promotion	1,200.00
Prepaid Insurance Premiums—New York	193.25
Prepaid Insurance Premiums—Cleveland	724.26

Permanent Assets

The land and buildings, instruments, equipment and furniture and fixtures are reflected on the balance sheet at cost of acquisition. With the exception of land and buildings all assets have been depreciated at the rate of 10 percent per annum. During the year fully depreciated furniture and fixtures in the amount of $287.63 was written off against the reserve.

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949 depreciation was not provided on the buildings for the current fiscal year.

Taxes

The sum of $552.70 represents Federal income taxes withheld from employees' salaries during the month of October, 1950.

Accrued Accounts

The salaries and commissions accrued as of October 31, 1950 have been computed by us in accordance with the Finance Committee report of April 8, 1948. Other accruals cover various expenses applicable to the current fiscal year.

Deferred Income

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follows:

Membership	Amount
Members	$ 6,263.00
Associates	4,268.00
Affiliates	314.14
Juniors	1,094.34
Students	78.00
Total	$12,017.48

The balance sheet of the Society also reflects as deferred income, balances in the various earmarked funds unexpended as at October 31, 1950.

Reserve for Transactions

Out of the sum of $12,000.00 budgeted for the publication of Volume 54 there remains a balance of $3,646.94 which is considered ample for the completion of this publication. In conformity with the 1949 and 1950 budgets reserves have been provided for the publication of Volumes 55 and 56.

FUNDS

There is included as a part of this report an analysis of funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1950.

INSURANCE

The insurance coverages of the Society follow:

Fire—Buildings (Cleveland)		$151,000.00
Fire—Personal Property:		
Cleveland	$19,000.00	
New York City	20,000.00	
Printers	1,000.00	40,000.00
Sprinkler Leakage		11,000.00
Comprehensive—Bodily Injury		100/300M
Comprehensive—Property Damage—Automobiles		5,000.00
Parcel Post Open Policy		600.00
Commercial Blanket Bond on all Employees including the President, Treasurer and Chairman of the Finance Committee		20,000.00
Workmen's Compensation		

Respectfully submitted,

FRANK G. TUSA & CO.
Certified Public Accountants

BALANCE SHEET

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.
October 31, 1950

ASSETS

GENERAL FUND				
CASH				
On Hand and on Deposit			$84,762.91	
SECURITIES				
At Cost (Market Value $38,400.97)		$36,110.75		
Add: Accumulated Interest		2,267.05	38,377.80	
ACCOUNTS RECEIVABLE				
Membership Dues	$15,809.01			
Less: Reserve for Doubtful	12,000.00	3,809.01		
Advertisers and Sundry Debtors	6,294.13			
Less: Reserve for Doubtful	1,000.00	5,294.13		
Initiation Fees		77.00	9,180.14	
INVENTORIES			14,679.96	
DEPOSIT WITH UNITED AIRLINES			425.00	
ADVANCES TO EMPLOYEES PENSION TRUST			1,433.46	
DEFERRED CHARGES			3,943.86	$152,803.13
RESEARCH GENERAL FUND				
CASH				
On Hand and on Deposit			18,311.72	
ACCOUNTS RECEIVABLE				
40% of Dues	5,449.00			
Less: Reserve for Doubtful	4,000.00	1,449.00		
Due from U. S. Navy Department		8,994.32	10,443.32	

ADVANCES TO EMPLOYEES			985.77	
DEFERRED CHARGES			724.26	30,465.07
PROPERTY FUND				
Land and Buildings		84,920.96		
Less: Reserve for Depreciation		3,908.07	81,012.89	
Laboratory and Equipment (Cleveland)		50,157.89		
Less: Reserve for Depreciation		18,693.93	31,463.96	
Furniture and Fixtures—(New York)		8,555.20		
Less: Reserve for Depreciation		2,065.76	6,489.44	
Library			300.00	
Tools			300.00	119,566.29
RESERVE FUNDS				
Cash on Deposit			27,706.69	
Securities at Cost (Market Value $61,322.75)		53,631.50		
Add: Accumulated Interest		7,691.25	61,322.75	89,029.44
ENDOWMENT FUNDS				
F. PAUL ANDERSON FUND				
Cash		138.47		
Securities	1,000.00			
Add: Accrued Interest	12.50	1,012.50	1,150.97	
RESEARCH				
Cash on Deposit			674.10	1,825.07
				$393,689.00

LIABILITIES

GENERAL FUND				
FEDERAL WITHHOLDING TAX			$ 552.70	
DUE RESEARCH			1,449.00	
ACCRUED ACCOUNTS				
Salaries and Commissions		$ 9,250.15		
Expenses		2,347.51	11,597.66	
DEFERRED INCOME				
Prepaid Membership Dues:				
Elected Members	$12,017.48			
Candidates	715.00	12,732.48		
Prepaid Admission Fees		1,753.00		
Past President's Award		100.00	14,585.48	
RESERVE FOR PUBLICATIONS				
TRANSACTIONS—Vol. 54		3,646.94		
TRANSACTIONS—Vol. 55		12,500.00		
TRANSACTIONS—Vol. 56		12,500.00	28,646.94	$ 56,831.78
RESEARCH GENERAL FUND				
DEFERRED INCOME				
Sound Energy Studies			574.39	
Summer Infiltration			500.00	
Panel Heating Studies			5,582.50	
Odor Studies			2,000.00	
Heat Flow Through Roofs			500.00	9,156.89

FUNDS		
General Fund	95,971.35	
Research General Fund	21,308.18	
Property Fund	119,566.29	
Reserve Fund	67,776.98	
Research Reserve Fund	21,252.46	
F. Paul Anderson Fund	1,150.97	
Endowment for Research Fund	674.10	327,700.33
		$393,689.00

CONDENSED COMPARATIVE BALANCE SHEET

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New York, N. Y.

October 31, 1950

ASSETS	October 31, 1950	October 31, 1949	Increase Decrease
Cash on Hand and on Deposit	$131,593.89	$124,703.13	$ 6,890.76
Securities (At Cost—Plus Accumulated Interest)	100,713.05	95,392.19	5,320.86
Accounts Receivable (Less: Reserve for Doubtful)	18,174.46	21,396.43	(3,221.97)
Inventory of Paper, Emblems, Etc.	14,679.96	4,670.68	10,009.28
Deposits Receivable	425.00	437.00	(12.00)
Accommodations	2,419.23	2,784.23	(365.00)
Land and Buildings (Less: Reserve for Depreciation)	81,012.89	81,012.89	—0—
Laboratory Equipment, Tools, Furniture, Fixtures and Library (Less: Reserve for Depreciation)	38,553.40	26,984.43	11,568.97
Deferred Charges	4,668.12	5,087.30	(419.18)
TOTAL ASSETS	$392,240.00	$362,468.28	$29,771.72
LIABILITIES			
Accounts Payable	$ —0—	$ 118.75	$(118.75)
Federal Withholding Taxes	552.70	465.60	87.10
Accrued Accounts	11,597.66	12,208.47	(610.81)
Deferred Income:			
Research Projects	9,156.80	18,398.14	(9,241.25)
Prepaid Dues, Initiation Fees, Etc.	14,585.48	4,328.19	10,257.29
Reserve for Publication of TRANSACTIONS	28,646.94	26,151.85	2,495.09
TOTAL LIABILITIES	$ 64,539.67	$ 61,671.00	$ 2,868.67
FUNDS			
Property Funds	$119,566.29	$107,997.32	$11,568.97
Reserve Funds	67,776.98	80,718.10	(12,941.12)
F. Paul Anderson Fund	1,150.97	1,123.58	27.39
Endowment for Research Fund	674.10	659.17	14.93
Research Reserve Fund	21,252.46	30,560.06	(9,307.60)
General Fund	117,279.53	79,739.05	37,540.48
NET WORTH	$327,700.33	$300,797.28	$26,903.05
TOTAL LIABILITIES AND FUNDS	$392,240.00	$362,468.28	$29,771.72

STATEMENT OF INCOME AND EXPENSES

The American Society of Heating and Ventilating Engineers
New York, N. Y.
For the fiscal year ended October 31, 1950

INCOME

Income from Dues				
104—Members	$ 75,281.58			
Less: Cancellations	562.50	$ 74,719.08		
Less: 40% to Research Fund		29,887.63	$ 44,831.45	
105—Associates	72,830.17			
Less: Cancellations	1,245.00	71,585.17		
Less: 40% to Research Fund		28,634.07	42,951.10	
105A—Affiliates		2,375.00		
Less: 40% to Research Fund		950.00	1,425.00	
106—Juniors		11,674.00		
Less: Cancellations		295.00	11,379.00	
107—Students		2,814.00		
Less: Cancellations		123.00	2,691.00	
Total Dues				$103,277.55
Other Income				
108—Admission Fees			6,345.50	
109—Sales of Emblems			648.25	6,993.75
Income from Publications				
115—Editorial Contract			23,500.00	
300—Guides (Per Schedule Attached)			127,710.56	
117—Sales of Transactions			893.55	
118—Books, Reprints and Codes			1,832.14	153,936.25
Investment Income				
125—Interest—Savings Banks			382.91	
126—Interest—Securities			1,042.47	1,425.38
Research				
From A.S.H.V.E.				
40% of Members and Associates Dues				$ 56,391.46
From U. S. Navy Contract				25,504.90
Contributions General—Per Schedule			$ 6,445.00	
Earmarked				
Received during the Current Fiscal Year—Per Schedule	$ 46,709.29			
Add: Deferred from Prior Years	18,398.14	$ 65,107.43		
Deduct: Deferred to Future Operations		9,156.89	55,950.54	
1950 Exposition Contribution			15,757.13	78,152.67
Interest Earned				10.39
				$160,059.42
Total Income				$425,692.35

EXPENSES

COMMITTEES AND CHAPTERS—SCHEDULE F			
150—President's Fund	2,666.01		
151—Council Travel and Council Committee Expense	6,187.98		
170—Admissions and Advancement Committee Travel	540.74		
171—Constitution and By-Laws	195.61		
172—Nominating Committee Travel	996.69		
173A—Chapter Speaker Travel	2,497.06		
173B—Chapter Delegates Travel	4,077.97		
	17,162.06		425,692.35
173C—Chapter Relations Committee	360.69		
201—ASA Membership	100.00		
204—Membership Certificates	1,318.93		
205—Emblems	529.62		
206—Medals and Awards	172.80	19,644.10	
MEETING—SCHEDULE G			
163—Meetings	5,142.12		
327—Chapter Annual and Semi-Annual Meeting Allowances	1,682.81	6,824.93	
PUBLICATION—SCHEDULE H			
200—Members Subscriptions to HPAC	15,185.50		
202—TRANSACTIONS—Volume 56	12,500.00		
203—Membership Roll	6,074.64		
108—Books, Reprints and Codes	1,636.66		
305—GUIDE—Publication and Distribution (Per Schedule Attached)	70,208.82	105,605.62	
HEADQUARTERS—SCHEDULE I			
210—Salaries—Secretary and Staff	54,767.82		
211—Public Relations and Research Promotion	6,799.37		
212—Traveling—Secretary and Staff	2,936.57		
213—Rent and Light	5,947.49		
214—Telephone	1,953.17		
215—Telegraph	448.23		
216—Postage	5,008.86		
217—Printing and Stationery	6,802.05		
219—Addressing and Address Changes	552.70		
220—Professional Services	4,092.65		
221—Bank Charges	50.10		
222—Insurance	163.64		
11—Depreciation of Furniture and Fixtures	797.22		
223—Office Expense	1,676.50		
224—Pension	1,279.46	93,275.83	
RESEARCH			
CHAIRMAN AND COMMITTEES			
Travel and Meeting Expenses	1,855.19		
Chairman's Office Expenses	421.99	2,277.18	
STAFF SALARIES, ETC.			
Administrative and Technical Staff	69,177.33		
Clerical Staff	11,589.05		
Pension Contributions	1,268.07		
Temporary and Part Time	3,559.58		
Consulting Services	5,557.20	91,151.23	
LABORATORY EXPENSES			
Staff Travel	6,026.53		
Postage	678.08		
Telephone and Telegraph	1,953.68		
Office Supplies	1,074.42		

Mimeographing and Printing	2,114.44		
Library	525.25		
Accounting Fees	425.61		
Staff Insurance	189.61		
Laboratory Materials and Supplies	8,344.27		
Bank Charges	16.09		
Depreciation of Equipment and Fixtures	4,257.30	25,605.28	
		119,033.69	160,059.42
LABORATORY OPERATION AND MAINTENANCE			
Heat	1,111.88		
Utilities	1,752.20		
Janitor's Wages and Supplies	2,506.29		
Building Maintenance	1,341.54		
Building Insurance	943.77		
Building Alterations	259.20	7,914.88	
NAVY CONTRACT			
Salaries	10,669.11		
Travel (Committee)	209.36		
Travel (Staff)	239.47		
Materials and Supplies	7,562.97	18,680.91	
COOPERATIVE RESEARCH AGREEMENTS			
Case Institute of Technology	2,250.00		
Columbia University	800.00		
University of Minnesota	4,200.00		
University of Illinois	6,419.29		
Kansas State College	3,600.00		
Michigan State College	500.00		
University of Florida	1,000.00	18,769.29	164,398.77
		168,398.77	393,749.25
			31,943.10

NET INCOME

OTHER DEDUCTIONS			
Admission Fees Allocated to Reserve Fund		6,345.50	
Provision for Dues Doubtful of Collection		7,276.12	
Provision for Accounts Receivable Doubtful of Collection		747.70	
Loss Due to Difference in Canadian Rate of Exchange		102.98	14,472.30
EXCESS OF INCOME OVER EXPENSES			$ 17,470.80

BUDGET COMPARISON—CASH RECEIPTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.
For the fiscal year ended October 31, 1950

DUES—SCHEDULE A	Actual	Budget Provision	Increases	Decreases
CURRENT MEMBERS				
6A—Members	$ 39,700.12	$ 45,000.00	$	$ 5,299.88
6B—Associates	34,894.39	42,000.00		7,105.61
6BB—Affiliates	796.61	—0—	796.61	
6C—Juniors	7,727.49	8,000.00		272.51
6D—Students	1,934.00	1,800.00	134.00	
	85,052.61	96,800.00	930.61	12,678.00

NEW MEMBERS				
104—Members	1,762.65	1,875.00		112.35
105—Associates	2,142.15	1,875.00	267.15	
—Affiliates	291.00	—0—	291.00	
106—Juniors	1,338.50	500.00	838.50	
107—Students	—0—	150.00		150.00
	5,534.30	4,400.00	1,396.65	262.35
PRIOR YEARS DUES				
6E—Members	1,195.45	600.00	595.45	
6F—Associates	1,599.23	900.00	699.23	
6G—Juniors	219.50	200.00	19.50	
6H—Students	16.00	90.00		74.00
	3,030.18	1,790.00	1,314.18	74.00
TOTAL DUES	93,617.09	102,990.00	3,641.44	13,014.35
108—Admission Fees	5,113.50	4,000.00	1,113.50	
DEFERRED INCOME				
6I—Prepaid unbilled Dues	1,258.83	500.00	758.83	
—Prepaid billed Dues	10,712.15	—0—	10,712.15	
6J—Prepaid Dues by Candidates	715.00	1,000.00		285.00
108A—Prepaid Admission Fees	1,753.00	1,000.00	753.00	
	14,438.98	2,500.00	12,223.98	285.00
PUBLICATIONS—SCHEDULE B				
115—Journal Contract	23,500.00	23,500.00	—0—	—0—
300—GUIDE Advertising	71,014.15	58,000.00	13,014.15	
302—GUIDE Copy Sales	58,378.14	52,500.00	5,878.14	
117—TRANSACTIONS	901.09	1,000.00		98.91
118—Books, Reprints and Codes	1,994.17	500.00	1,494.17	
	155,787.55	135,500.00	20,386.46	98.91
INVESTMENTS—SCHEDULE C				
125—Interest—Savings Bank	381.72	300.00	81.72	
126—Interest—Securities	668.17	500.00	168.17	
	1,049.89	800.00	249.89	
OTHER RECEIPTS—SCHEDULE D				
109—Sale of Emblems	290.00	500.00		210.00
Exposition Fund	—0—	12,500.00		12,500.00
	290.00	13,000.00		12,710.00
RESEARCH LABORATORY RECEIPTS—SCHEDULE E				
40% of Members and Associate Dues	55,709.51	61,500.00		5,790.49
Contributions—Earmarked	46,709.29	60,000.00		13,290.71
Contributions—General	6,445.00	10,000.00		3,555.00
Contributions—Exposition	15,757.13	—0—	15,757.13	
Navy Research	18,347.59	37,500.00		19,152.41
Interest	559.68	—0—	559.68	
	143,528.20	169,000.00	16,316.81	41,788.61
TOTAL RECEIPTS	$413,825.21	$427,790.00	$ 53,932.08	$ 67,896.87

BUDGET COMPARISON—CASH DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS — New York, N. Y.

For the fiscal year ended October 31, 1950

COMMITTEES AND CHAPTERS—SCHEDULE F	Actual	Budget Provision	Increases	Decreases
150—President's Travel	$ 2,384.96	$ 3,000.00	$	$ 615.04
151—Council Travel and Committee Expense	6,625.57	7,500.00		874.43
170—Admission and Advancement Committee	540.74	750.00		209.26
171—Constitution and By-Laws	195.61	100.00	95.61	
172—Nominating Committee	996.69	1,500.00		503.31
173A—Chapter Speakers	2,025.79	2,000.00	25.79	
173B—Chapter Delegates	4,077.97	4,500.00		422.03
173C—Chapter Relations Committee	352.02	600.00		247.98
201—ASA Membership Dues	100.00	100.00		
204—Membership Certificates	1,189.60	750.00	439.60	
205—Emblems	216.62	1,000.00		783.38
206—Medals and Awards	172.80	250.00		77.20
	18,878.37	22,050.00	561.00	3,732.63
MEETINGS—SCHEDULE G				
163—Meetings	5,142.12	5,500.00		357.88
327—Chapter Meeting Allowance	1,682.81	2,000.00		317.19
	6,824.93	7,500.00		675.07
PUBLICATIONS—SCHEDULE H				
200—Journal Subscriptions	15,089.40	15,000.00	89.40	
202—TRANSACTIONS—Volumes 54 and 55	8,688.47	15,000.00		6,311.53
TRANSACTIONS—Volume 56		12,500.00		12,500.00
203—Membership Roll	6,074.64	5,000.00	1,074.64	
208—Books, Reprints, Codes	1,727.08	500.00	1,227.08	
305—325—GUIDE Publications and Distribution	80,673.86	67,500.00	13,173.86	
	112,253.45	115,500.00	15,564.98	18,811.53
HEADQUARTERS—SCHEDULE I				
210—Salaries	54,639.82	67,600.00		12,960.18
210A—Social Security Tax		500.00		500.00
211—Public Relations and Research Promotion	6,820.29	8,500.00		1,679.71
212—Travel, Secretary and Staff	2,936.57	3,000.00		63.43
213—Rent and Light	5,825.36	8,850.00		3,024.64
214—Telephone	1,845.11	1,500.00	345.11	
215—Telegraph	571.93	500.00	71.93	
216—Postage	5,015.64	4,500.00	515.64	
	77,654.72	94,950.00	932.68	18,227.96
217—Printing and Stationery	6,777.12	5,000.00	1,777.12	
219—Addressing and Address Changes	407.60	500.00		92.40
220—Professional Services	4,892.65	3,000.00	1,892.65	
221—Bank Charges	50.10	500.00		449.90
222—Insurance	130.29	500.00		369.71
11—Furniture and Fixtures	1,453.66	1,500.00		46.34
223—Office Expense	1,641.40	1,500.00	141.40	
224—Pension	1,279.46	1,500.00		220.54
	94,287.00	108,950.00	4,743.85	19,406.85

RESEARCH—SCHEDULE J				
Research Committee Expenses	2,277.18	2,500.00		222.82
Staff Salaries	91,151.23	98,450.00		7,298.77
Laboratory Expenses	36,624.56	45,050.00		8,425.44
Laboratory Operation and Maintenance	7,133.00	9,000.00		1,867.00
Cooperative Contracts	18,769.29	22,500.00		3,730.71
Navy Research	18,680.91	26,250.00		7,569.09
	174,636.17	203,750.00		29,113.83
TOTAL DISBURSEMENTS	$406,879.92	$457,750.00	$ 20,869.83	$ 71,739.91

There were three technical papers presented at the first session (see program p. 32). A resolutions committee was appointed at the close of the session by President Avery with instructions to report at the final session, consisting of S. W. Beaty, Chairman, Shreveport, La.; R. C. Chewning, Portland, Oregon; I. W. Cotton, Indianapolis, Indiana.

SECOND SESSION, TUESDAY, JANUARY 23, 9:30 A.M.

First Vice-Pres. L. E. Seeley presided at the second session and called on H. K. Jennings of the Board of Tellers of Election for his report.

REPORT OF TELLERS OF ELECTION

BALLOT FOR OFFICERS	Total
President, L. E. Seeley, Durham, N. H.	1837
1st Vice Pres., Ernest Szekely, Milwaukee, Wis.	1837
2nd Vice Pres., Reg F. Taylor, Houston, Tex.	1837
Treasurer, H. E. Sproull, Cincinnati, Ohio	1840
MEMBERS OF COUNCIL (*Three-Year Term*)	
J. H. Fox, Toronto, Ont., Canada	1837
L. N. Hunter, Johnstown, Pa.	1840
N. H. Peterson, San Francisco, Calif.	1838
B. H. Spurlock, Jr., Boulder, Colo.	1839
Total Ballots Received	1842
Total Legal Ballots	1841
Invalid Ballots	1
Scattering votes:	
President	4
1st Vice President	4
2nd Vice President	4
Treasurer	1
Council	7
Committee on Research	3
COMMITTEE ON RESEARCH (*Three-Year Term*)	
A. B. Algren, Minneapolis, Minn.	1838
John Everetts, Jr., Philadelphia, Pa.	1839
T. H. Smoot, Marshalltown, Ia.	1839
T. A. Walters, Detroit, Mich.	1839
W. N. Witheridge, Detroit, Mich.	1837

RESPECTFULLY SUBMITTED,

H. K. JENNINGS, *Chairman*
E. A. DUSOSSOIT
J. S. KEARNEY

After two technical papers were presented, R. C. Cross, Chairman of the Committee on Research, gave his report which follows:

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1950

IT IS a pleasure, and a privilege to have this opportunity to present to the members of the Society a report of the Committee on Research for the past year of operation. Our Society is unique in the respect that it is the only professional Society which supports its own Research Laboratory, and the research program very properly should be of interest to each and every member of the Society. It is the proportionate allotment of membership dues that supports the research program to a large part and because of this, the Committee on Research feels that it has a very definite obligation to the members as well as to the Society in general and to the art of heating, ventilating, and air conditioning.

Consistent with the general development of the industry and the increasing membership of the Society, it might be well to note the increase in the dollar volume for research expenditures from a low of approximately $32,000 in 1942 to last year's figure of $160,000.

Your attention is called to the increase in projects over the past 10 years from a low of 13 projects in 1943 to 35 projects in 1950. The laboratory staff has increased from a low of 5 members to its present complement of 25 persons. These data are presented as a general review of progress during the past 10 years, and would indicate a normal, healthy growth.

During the year 1950, the research program has been actively coordinated by 19 Technical Advisory Committees. Twenty-five Technical Advisory Committee meetings were held during the year with 189 members present, and 120 visitors attending. The Committee on Research held three meetings with 42 members and six visitors in attendance. Three meetings of the Research Executive Committee were held. All of this activity is indicative of the interest and enthusiasm of the members of these committees, and when one considers the fact that the services of these men are purely voluntary, it is the debt of the general membership to these groups which must be gratefully acknowledged.

The following projects were completed at the Research Laboratory during the year 1950.

1. Measurements of heat flow through unshaded sunlit glass.
2. Development of design heat flow data, including over-all coefficients, for unshaded sunlit glass for three latitudes and one season.
3. Revision of THE GUIDE Chapter on Cooling Load to include above data.
4. Investigation of convection conductances for a flat surface exposed to the natural wind.
5. Development of new psychrometric chart and its inclusion in THE GUIDE 1951.
6. Development of simple heated thermocouple anemometer.
7. Survey and analysis of available information on energy losses in 90 deg duct elbows.
8. Revision of THE GUIDE Chapter on Air Duct Design to include data from above.
9. Field studies of heat losses from concrete floor panels.
10. Laboratory studies of the thermal characteristics of four plaster panels.
11. Studies, by electrolytic analogue, of heat flow through a concrete slab on earth with tubes tangent to the slab-earth interface.
12. Investigation of the Combustion Reference Test Unit as a means of rating fuel oils (in cooperation with the *Oil-Heat Institute*).
13. Measurement of fuel oil smoke density as part of the instrumentation for the Combustion Reference Test Unit.

In addition, two cooperative projects were completed:

Case Institute of Technology—Comparative study of ventilating jets from various types of outlets.
Columbia University—Electric Analogger studies on panels with imbedded tubes.

At the present time, there are 23 projects in progress at the Laboratory related to the following fields of interest:

Solar Heat Gain and Shading Effects

1. Investigation of variation of diffuse solar radiation incident upon variously oriented vertical surfaces.
2. Relationship of direct and diffuse irradiation of horizontal surfaces and solar altitude.
3. Analytical and experimental investigation of the effects of various types of sunshades on heat flow through sunlit glass.
4. Film coefficients for low air velocities.
5. Studies with a convection-compensated radiometer.
6. Preparation of bulletin summarizing research on heat flow through glass.

Panel Heating and Cooling

7. Electrolytic analogue studies of concrete panels.
8. Field studies of heat losses from concrete floor panels.
9. Field studies of heat flow from plaster panels.
10. Completion of calorimeter room and installation of equipment.
11. Development and construction of special instrumentation for calorimeter room.
12. Study of the nature and quantitative value of heat exchange from a floor or ceiling panel with temperature of all other surfaces controlled.

Air Distribution

13. Completion of literature survey on the fundamentals of ventilation jet behavior.
14. Study of isothermal jets discharging into a large space.
15. Investigation of turbulence in isothermal air movement.
16. Development of turbulence intensity indicator.
17. Development of probe for remotely establishing direction of air velocity.

Air Cleaning

18. Study of factors influencing development of standard methods for testing air cleaning devices.

Odors

19. Planning of odor-free test rooms and development of program for study of relationship of principal physical characteristics of air to development and perception of odors of varying types and levels.

Cooling Load

20. Development of thermal circuit diagram to give schematic representation of factors that influence the instantaneous load in an air conditioning system.

Heating Load

21. Completion of apparatus for measurement of infiltration or air change by tracer gas technique.

Sorbents

22. Field studies to check applicability of proposed method for testing sorption-type dehumidifiers.

Human Calorimetry (O.N.R. Contract)

23. Continuation of assembly, construction and instrumentation of component parts of human calorimeter and auxiliary equipment.

It has been a practice for some years for the Society to round out the research program beyond the limitations of the Research Laboratory by the conduct of research projects in cooperation with institutions of learning. It is felt that this is a healthy program, and the Committee on Research for the past year has attempted to strike

a balance between the work in progress at Cleveland and that being conducted elsewhere. There are nine cooperative projects now under way. The institutions and the projects are listed in the Director's report.

The policy of carrying on certain projects by means of Society sponsored fellowships at Universities has been reactivated during the past year to a certain extent. At the moment, fellowships have been established at the University of Minnesota and at Pennsylvania State College.

Consistent with the expansion of the program and the normal growth of the Laboratory, there have been significant additions to the Laboratory equipment in Cleveland. Those who have had the opportunity to visit the Laboratory during the past year have seen this in detail, but I would point out several of the more outstanding additions that have been made.

One of the most useful and valuable additions to the facilities is the new calorimeter room, which is well under way and will be available for active use within the near future. Essentially this is a room 25 ft × 12 ft with an adjustable ceiling in heights up to 12 ft. The room, with auxiliary equipment, consisting of compressors, a silica gel dehumidifier, heat exchangers, and accessory equipment and instrumentation will represent an investment of approximately $50,000. The Society is grateful to many friends in industry for donations of equipment or for providing equipment at favorable prices with prompt delivery, which expedited the building of this facility.

Another item of considerable interest and value to the program is the human calorimeter which has been constructed in connection with the program for the Office of Naval Research.

In addition to the supervision of the research program during the past year the Director of Research has assisted in fund raising activities and research promotion efforts. Although no special appeals have been made and booklets issued as in previous years, the Society has received contributions for the fiscal year amounting to approximately $53,000. I should like to take this opportunity to acknowledge Mr. Tasker's part in these efforts.

Although not directly a part of the 1950 research program, it should be of interest to the members to learn that, after considerable discussion of possible methods of raising funds for research, a special committee of Council, a Ways and Means Committee, has been appointed. This committee will assume the responsibility of fund raising under the general guidance of the Finance Committee of Council.

Considerable effort was expended during the fiscal year 1950 by the Committee on Research, and particularly by a sub-committee headed by I. W. Cotton, on the formulation of a long-range research program. The establishment of such a program is essential for presentation to the group assigned the responsibility of raising funds for research. It seems reasonable to presume that in order to raise funds for research properly, a well planned program must be presented to prospective donors. We should ever keep in mind our obligation to the membership, to industry, and to the art of maintaining the highest quality in the research program, and keep within the bounds of a carefully planned and balanced program.

The critical international situation that confronts us today has already shown indications of its possible effects on both the staff and the program. The services of the Society's research staff were made available to, and used by the nation in World War II. It is hoped that there will be opportunities to put our particular skills and knowledge to use for the common good once again.

It has been the practice in the past, and this report may be considered representative, for the Committee on Research to report officially to the members at the Annual Meeting of the Society. Some thought is being given to the desirability of more frequent reports of the status of the program, possibly through the Society JOURNAL.

In 1949 the membership was polled as to its interests and desires for projects to be considered in the research program. This information has been most helpful in guiding the deliberations of the Committee on Research for the past two years. It

may be desirable that this be done at periodic intervals in order that the reaction of the members to the program itself, and to advances in the art may come to the Committee on Research without undue delay. This matter, likewise, is being considered.

The solicitation of your ideas merely substantiates the belief that this is *your* Society, and this is *your* program supported by *your* money. It has been the aim of the Committee on Research during the past year to meet your ideas to the best of its ability.

In closing, I would like to pay tribute to the other members of the Committee on Research and to the numerous members of the Technical Advisory Committees, who have given so untiringly of their time and effort in the past year's program. Their invaluable contribution is deeply and gratefully acknowledged.

Finally, I wish to pay tribute to Cyril Tasker, our director of research, whose enthusiasm, purpose, and grasp of the situations arising in the administration of the research program have been a major contribution to whatever success the research program may have achieved in the year 1950.

REPORT OF THE DIRECTOR OF RESEARCH

As reported by Cyril Tasker, Director of Research, the year 1950 was marked by continual activity. For most of the year, more projects were on the program than there was competent staff in Cleveland to attack them or qualified institutions able and willing to assist. The Staff strove to maintain the quality of the program and to resist the temptation to replace quality with quantity.

The record for 1950 should not be measured only by the papers presented at the 1950 Semi-Annual Meeting and at the 57th Annual Meeting. The results of much of the work will come to light when THE GUIDE 1951 is published.

EQUIPMENT

The growing dollar value of our inventory of apparatus and equipment is a poor yardstick by which to evaluate it. Most of the apparatus and equipment now in use has a high *use factor* and expensive pieces of equipment are moved as needed from project to project.

No major improvements were made to the property in 1950 but it is being maintained in good condition.

STUDIES AT CLEVELAND LABORATORY AND COOPERATING INSTITUTIONS

Brief reviews of the projects underway at the Cleveland Laboratory and at the nine institutions cooperating in the 1950 program are arranged under a number of general headings. Fuller details are given in minutes of the Technical Advisory Committees and in progress reports. Visits of Society members to the Laboratory and visits of staff members to organizations within the industry, to our technical meetings, and to meetings of other Societies and associations help to maintain that close liaison between the research staff and industry that is so important. It has been the writer's privilege to be able to make more personal contacts in 1950 than in previous years and to gain thereby a better appreciation of the part that the Society's research work can play in the future of this growing industry.

AIR FLOW AND DISTRIBUTION

Studies coming under the general direction of the TAC on Air Distribution continued to occupy an important place in the program.

Air Flow: At the Laboratory, D. W. Locklin completed an extensive survey and analysis of information available in the American and foreign technical press on

energy losses in 90-deg duct elbows. This method of approach, yielded empirical correlations and data which are quite satisfactory for most engineering uses. The chapter on Air Duct Design for THE GUIDE 1951 has been revised to include these data. The survey should be extended to cover losses in vaned elbows, compound elbows and elbows involving change of area.

At Michigan State College, under Lorin G. Miller, studies on the resistance to the flow of air offered by various types of take-off fittings from a rectangular main have continued. These studies present a number of difficult experimental problems, as well as the problem of presenting the results in a simple usable form.

Room Air Distribution: Studies have continued at the laboratory, at Case Institute of Technology, and at Kansas State College. The work at the Laboratory, under H. B. Nottage, has for its objective the establishment of reliable data, both fundamental and practical, on the movement of air arising from the introduction of ventilating jets into confining spaces under non-isothermal conditions. Variables include the size and arrangement of the confining space, the volume of air handled, the initial jet velocity, the turbulence intensity and the temperature difference relative to the surrounding air. Special equipment has been provided and a fundamental attack is being made on these problems. A comprehensive literature survey on the fundamentals of ventilation jet behavior is being prepared.

As part of the instrumentation for these studies, a new type of heated-thermocouple anemometer was developed and the methods of construction and use reported in a paper[1] at the June Meeting. The staff working on this project have also developed a probe for remotely establishing the direction of air velocity. A device to measure turbulence intensity was developed and is in regular use; details will be published later. Smoke trail techniques for determining velocity directions at extremely low velocities have been perfected and used in this work.

Case Institute of Technology—G. L. Tuve and his colleagues presented the results[2] of a comparative study of ventilating jets from various types of outlets and showed that a single simple method could be used in most cases. Later, A. Koestel and Chia-Yung Young, both of Case Institute of Technology, submitted a paper on *The Control of Air Streams from a Long Slot.* It presented a method of obtaining uniform air discharge along the entire length of a long slot in the side of a supply duct. Theoretical flow equations were developed and verified by experiment and a practical means of application suggested.

Current studies are concerned with the investigation of annular slots, ceiling plaques and circular diffusers.

Kansas State College—Studies on the downward projection of heated air are being carried on under Linn Helander with the cooperation of the *Industrial Unit Heater Association.* During the year tests have been made of the maximum blow from standard *A.S.M.E.* nozzles, 8 in., 10 in., 12 in., 14 in., and 17 in. in diameter with initial velocities ranging from 13 to 53 fps and initial temperatures ranging from 156 F to 264 F. From the data secured, a formula for maximum blow has been developed which it is expected will provide data of practical value for design purposes over a fairly wide range. The studies are continuing.

PHYSIOLOGICAL RESEARCH

Human Calorimetry (O.N.R. Contract): The 1949 report gave details of the design of the calorimeter and other component parts. At the year end the calorimeter shell had been assembled, and some of the piping completed; equipment for the flow circuits for the calorimeter and the ventilatory air had been delivered. A great deal of work had also been completed on the heat metering layers.

The original contract with the O.N.R. expired on June 19, 1950 and was replaced with a new two-year contract.

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[1] A.S.H.V.E. Research Report No. 1402—A Simple Heated-Thermocouple Anemometer, by H. B. Nottage (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 431).

[2] A.S.H.V.E. Research Report No. 1404—Comparative Study of Ventilating Jets from Various Types of Outlets, by Alfred Koestel, Philip Hermann, and G. L. Tuve (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 459).

Environmental Effects: Studies at the College of Medicine, University of Illinois, were continued. The research grant-in-aid of the *National Institutes of Health, U. S. Public Health Service,* was renewed for the period July 1950 to June 1951. Society funds were added to this grant.

These studies were planned to determine the existence or absence of physiologic strain in normal and abnormal subjects as shown by an alteration in energy requirement or initiation of a patho-physiologic process. The subjects were moved from a comfortable room to a hot room and back. Additional tests were made with healthy subjects to evaluate recently established ideas of strain.

The study of effects of summer weight clothing on the physiologic adjustments to movement from comfortable to hot environment and back has been completed on 8 normal young women subjects. No gross evidence of strain was obtained.

Work is in progress on other cases including obese and cardiac patients; patients with impaired circulatory mechanisms; and effects of humidity on healthy young men.

HEAT FLOW AND HEAT TRANSFER

Heat Transmission Through Glass: The planned experimental work on measurements of heat flow through unshaded sunlit glass of various types was virtually complete early this year. The paper[3] on solar energy transmittance of figured rolled glass, completes the reporting of the main details of these studies, which have been under way for several years. Several supplementary investigations were undertaken during the year. For example, a study was made of convection conductances for a flat surface exposed to the natural wind. The heat transfer surface was a single sheet of glass of known solar absorption characteristics. This was installed in the calorimeter, which supplied the heat. and was exposed to winds of various velocities. Measurements were made with the surface parallel to, perpendicular to, and sheltered from the wind flow. These data were necessary in determinations of solar absorptances from previously obtained test data on glass block and configurated glass.

In order to facilitate the development of design heat flow data for sunlit glass, a considerable number of observations were made of the variations of diffuse solar radiation incident upon variously oriented vertical surfaces. A graphical relationship was developed between these values, similar data for a horizontal plane, and the altitude of the sun. Analysis of solar data obtained during the past few years developed other useful relationships between direct and diffuse irradiation of horizontal surfaces and solar radiation.

The results of another investigation, made to determine the possible relationship between diffuse transmittance by a sheet of glass and its orientation with respect to the sun, seemed to indicate a systematic variation, with incident angle of the order of 10 percent between zero and 90 deg incidence.

A paper which summarized the Society's work in fields related to solar heating, particularly the glass heat-flow research, was presented by G. V. Parmelee to the M.I.T. Symposium on Space Heating by Solar Heat held in Boston in August 1950.

Design Data on Heat Flow Through Glass: Considerable progress was made in the development of usable design data. The paper[4] on heat flow through unshaded glass—design data for use in load calculations, gave design values of overall coefficients and considerable data on solar heat gain. The latter were greatly extended and prepared by the Laboratory staff for the chapter on Cooling Load in THE GUIDE 1951.

Steps are being taken to summarize in a research bulletin all additional design data as well as experimental data and other information heretofore unpublished.

Convection-Compensated Radiometer: Previous annual reports have mentioned the development of an instrument to measure incoming radiation from all sources. This

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[3] A.S.H.V.E. Research Report No. 1417—Solar Energy Transmittance of Figured Rolled Glass, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 209).

[4] A.S.H.V.E. Research Report No. 1399—Heat Flow Through Unshaded Glass—Design Data for Use in Load Calculations, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 371).

equipment has been in use for the greater part of 1950; many observations have been made. A paper describing the instrument and the application of data secured with it is now in preparation.

Shading Studies: An analytical approach to the evaluation of the effects of various types of sunshades was started in 1950 followed by experimental work in the summer and fall. For the present, emphasis has been placed on the slat type of shade. Tests have been conducted on slats of two colors, each at two different slat angles. These results are now being compared with the analytical treatment, which will be extended in the coming year. This comparison requires spectral reflectance data, which are being secured by a commercial laboratory. Tests for absorption of the total solar spectrum are being made at the Laboratory. It is anticipated that this approach to the very complex problem of shading will result in generalized solutions applicable to the large number of glass-shade combinations.

As in previous years, financial and technical support was given to these studies during 1950 by organizations in the glass industry and some segments of the air conditioning industry. This year some of the prominent organizations in the shading industry also supported these studies.

Cooling Loads—Periodic Heat Flow: C. O. Mackey is continuing his studies at Cornell University with the sol-air thermometer. A careful analysis of the data on sol-air temperatures for New Orleans, developed under a cooperative research contract with Tulane University, and the data previously published for New York City and Lincoln, Nebr., showed that it would be advisable to correlate all these data and get them onto a common footing.

At the Semi-Annual Meeting, R. H. Heilman and R. W. Ortmiller of *Mellon Institute of Industrial Research*, Pittsburgh, reported[5] the results of tests, made at the suggestion of the TAC on Cooling Load (W. E. Zieber, *chairman*), on the effective solar absorption of various colored paints.

Instantaneous Heat Gains: H. B. Nottage of the Laboratory staff has recently prepared, in preliminary form, a thermal-circuit diagram to attempt to portray and interrelate the major factors which influence the instantaneous thermal load imposed upon air conditioning equipment. This is being submitted for committee discussion and may serve as an outline for a possible analogue approach toward a solution of the time-variable nature of load constituents, a subject which has been given increasing attention in the air conditioning industry in recent years.

Heat Pump: During the year the the TAC on the Heat Pump (R. C. Jordan, *chairman*) completed a thorough review of various heat sources and sinks for heat pump applications and made a comprehensive outline of needed information and research that should be carried on to provide required data. A description of this program appears on page 87 of the November 1950 issue of *Heating, Piping & Air Conditioning*.

At the University of Minnesota under a cooperative contract, R. C. Jordan has continued the study of the availability of solar energy as a potential heat source. Phases under study at the present time include investigation of the theoritical availability of solar radiation, evaluation of recorded data, study of horizontal versus vertical radiation and an attempt to provide a rational procedure by which the availability of solar radiation in various geographical areas may be determined.

At Columbia University, under a cooperative research contract, W. A. Hadley and C. F. Kayan have been making an investigation of thermally actuated moisture migration in granular media. This problem was initiated when it was discovered that more heat was arriving at the ground coil of a heat pump than heat transfer theory would predict. Later work showed that there was a significant flow of moisture actuated by the thermal gradient and that its mode of transfer was apparently incompletely understood. Radioactive cobalt is being used as a moisture tracer and its passage is being followed by a Geiger counter.

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[5] Effective Solar Absorption of Various Colored Paints, by R. H. Heilman and R. W. Ortmiller (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 399).

PANEL HEATING AND COOLING

Studies under the general direction of the TAC on Panel Heating (P. B. Gordon, *chairman*) have been one of the major activities of the Laboratory during 1950. Work has also been carried on at one cooperating institution.

Two papers[6, 7] were published from the Laboratory studies. The paper, Electric Analogger Studies on Panels with Imbedded Tubes, reported the results of the studies made at Columbia University by Carl F. Kayan under a cooperative research contract.

Plaster Panels: An investigation into the thermal characteristics of plaster panels was completed and a report prepared for committee study.

The tests showed that the difference between the average tube temperature and the average panel surface temperature can be related to the heat output from the lower and upper panel surfaces by an empirical equation. The tests also showed the value of insulation back of a plaster panel and the effect of complete tube imbedment.

Field Tests on Concrete Floor Panels: During the winter 1949-1950, tests were conducted in four homes in Parma Heights, Ohio, to obtain data on the heat losses to the earth from floor panels and to determine the effectiveness of insulation in reducing these losses. A paper[8] reporting the results of the first season's tests was presented.

Analogue Tests on Concrete Panels: To extend the range of the experimental thermal tests on concrete panels, it was decided early in the year to modify the electrolytic analogue used in earlier work to permit the investigation of the slab resistance for the case of a concrete panel laid on earth with the tubes tangent to the earth-concrete interface. A complete report is being prepared for committee study.

Calorimeter Room: As indicated in the 1949 Report, the primary objective in building the calorimeter room was to provide facilities for studies of heat flow from heated or cooled panels to the space. However, the views of several other technical advisory committees were carefully considered in developing the final designs for this room. Construction commenced late in 1949 and by mid-February the TAC on Panel Heating, in consultation with the Laboratory staff, had decided on the main details of the construction. At the end of the year all major construction had been completed, the panels had been fitted, practically all items of equipment had been delivered and set in position, and piping was in progress. All construction, including the arrangements for adjusting the height of the ceiling, was carried out by members of the Laboratory staff at a cost much lower than the original estimates.

While the construction was under way, other staff members were making analytical studies of possible heat exchange in the room, to determine factors likely to be of significance in the planned programs, and also to aid in solving problems of instrumentation. The studies established the number and location of heat meters and of surface temperature thermocouples, and work began at once on the construction of some 150 low inertia heat flow meters that will meter heat exchanges at all room surfaces. The analyses have indicated that special attention must be directed to the evaluation of convection conductances at panel surfaces, as affected by position, air change rate and room surface temperature distribution.

Control of Panel Heating Systems: A research fellowship was set up at the University of Minnesota for field studies on the control of panel heating systems, in accordance with the recommendations of *Group D* (J. S. Locke, *chairman*) of the TAC on Panel Heating and Cooling.

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[6] A.S.H.V.E. Research Report No. 1387—Laboratory Studies on Heat Flow Within a Concrete Panel, by C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum, and D. W. Locklin, (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 175).

[7] A.S.H.V.E. Research Report No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 189).

[8] A.S.H.V.E. Research Report No. 1418—Field Studies of Heat Losses from Concrete Floor Panels, by C. M. Humphreys, C. V. Franks, L. F. Schutrum (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 221).

Five buildings have been selected for the first year's tests. Two of them, both offices, have heavy panels, heavy construction, and a large glass area. A manufacturing plant was chosen because it had a heavy panel, light construction, and a large glass area. The other two buildings are residences, one with a heavy panel, light construction, and small glass area, the other with a light panel, light construction, and a large glass area.

It is anticipated that the instrumentation will be completed in time to obtain data during the 1950-51 heating season.

The plan is to modify the control system in each building so that several general control methods can be used.

Guide Chapter: Under a subcommittee of the TAC on Panel Heating and Cooling, the chapter on panel heating was completely rewritten for THE GUIDE 1951.

Industry Support: Industry support of the panel heating research program during 1950 was very encouraging. A substantial part of the funds contributed by industry in 1949 was carried over to 1950 and added to the funds received during the year and to general funds available to the Committee on Research.

AIR CLEANING

It is now almost four years since active work began at the Laboratory on a program formulated by the TAC on Air Cleaning (R. S. Farr, *chairman*) and designed to meet an apparently urgent need for sound and acceptable codes for the testing and rating of air cleaning devices. From the first it was recognized that the problem was complex, and that, in spite of a great deal of work on the subject by competent investigators here and abroad, no standards had been established that were accepted equally by the filter industry and by large users. In the 1948 Annual Report of the Committee on Research, it was suggested that if the principal filter users could say what they wanted a filter to do, it might be of considerable help in defining both the methods and the degree of accuracy to be sought in filter testing. A review of the discussions at the U. S. Conference on Atmospheric Pollution in May 1950 would lead one to believe that the question has been answered only imperfectly, if at all.

The program so far has been designed to answer specific questions relating to the significance of certain factors in the development of standard methods of test. Among these are the velocity and dust distribution in the test duct; the importance of isokinetic sampling, *i.e.*, the removal of the sample at a rate that corresponds to the duct velocity; the location of the samplers; the number and type of samples to be taken; and the relationship between the size of the sampler intake and conditions within the duct, both before and after the filter. Of major importance is the question of the type or types of test dust to be used, and the range of particle sizes that it may be necessary to use in testing different types of air cleaning devices for different applications. The question that has always raised the most controversy is whether efficiencies shall be determined on a weight basis, a discoloration basis, or on both.

A number of reports have been made to the members of the Technical Advisory Committee over the past three years, and the program has followed the lines laid out by them.

PSYCHROMETRY

The steps leading to the presentation of a new psychrometric chart were outlined in the Report for 1949. H. B. Nottage presented the new chart[9] in a paper at the Semi-Annual Meeting. The new chart was accepted and it will appear, reproduced in two colors, in THE GUIDE 1951. A revision has also been made of the chapter on thermodynamics to bring it into line with the new chart. It has been decided to await the general reaction of Guide users to the new chart before proceeding with

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[9] A.S.H.V.E. Research Report No. 1401—A Proposed Psychrometric Chart, by H. B. Nottage (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 411).

the preparation of additional charts for pressures other than standard barometric and for higher and lower temperature ranges.

The work has been carried on in cooperation with the Guide Committee, which requested the Committee on Research to undertake this important project.

OIL-HEAT INSTITUTE

Following a meeting in February with the Engineering Committee of the *Oil-Heat Institute of America*, the results of the three investigations carried on in cooperation with the Institute were summarized in two papers[10, 11] presented at the 57th Annual Meeting.

The TAC on Combustion (T. H. Smoot, *chairman*) has suggested that the results obtained to date justify further investigation of the possibilities of the Combustion Reference Test Unit as a means for comparing fuel oils on the basis of their observed burning performance.

ODORS

Following a series of discussions by the TAC on Odors (T. H. Urdahl, *chairman*) and a review of possible methods of attack on the problem of odor perception and treatment in occupied spaces, the Committee on Research approved the recommendation that a preliminary survey of the odor problem be undertaken jointly by Arthur D. Little, Inc., Cambridge, Mass., and the Research Laboratory. The survey commenced on June 1 and was completed by mid-August. W. P. Gojsza, of the Laboratory staff, worked with three members of the survey section of Arthur D. Little in interviewing representatives of the heating, ventilating, and air conditioning industry regarded as well-informed on its odor problems. The report of the survey, submitted late in August, stated that the results indicated (1) that an odor problem of serious significance exists for the industry, and that substantial effort is warranted to solve it; (2) that tobacco, bodies, food and air conditioning equipment were considered to be the most troublesome and common odor sources; (3) that odor control was considered largely an economic problem; (4) that the establishment of an objective means of determining amounts of odors present is highly desirable, but there was considerable doubt that this could be attained by the Society within a reasonable period.

The TAC on Odors recommended that the Society undertake a research program on odors aimed initially at applied research, and that studies be commenced as rapidly as possible to investigate (*a*) the relationship of the principal physical characteristics of air, particularly temperature and humidity, to the development and perception of odors of varying types and levels; (*b*) the role of conventional air conditioning in odor reduction; and (*c*) the cause of air conditioning odors under various operating conditions with respect to the system components.

These recommendations were approved by the Committee on Research. Odor studies were included in the 1951 research program and initial steps taken to design and construct two relatively odor-free test rooms at the Cleveland Laboratory.

The report also proposed the development of a long-range program of fundamental research in odors, in cooperation with other interested groups. It was suggested that one of the first steps might be the holding of a National Symposium on Odors. At the year end preliminary negotiations towards this end were under way, in cooperation with certain divisions of the Office of Naval Research.

HOT WATER AND STEAM HEATING

A new Technical Advisory Committee (J. W. James, *Chairman*) was set up in mid-year to advise the Committee on Research regarding studies on problems in hot

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[10] A.S.H.V.E. Research Report No. 1412—Smoke Measurement in a Fuel Oil Test Unit, by D. W. Locklin and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 129).

[11] A.S.H.V.E. Research Report No. 1413—Rating of Fuel Oils by a Test Unit, by D. W. Locklin and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 139).

water and steam heating. One of the projects assigned to this Committee for advisory guidance was a cooperative research study on air entrainment in hot water heating systems, set up earlier in the year by the Committee on Research at the University of Florida.

At the organization meeting, the Committee recommended that a fundamental study be undertaken to determine the relationship between noise and pipe size, piping arrangement and fluid velocity in hot water and steam heating systems. Steps are being taken to set this up as a cooperative research project.

Sorbents

The TAC on Sorbents (John Everetts, Jr., *chairman*) developed, early in the year, a recommended method for testing and rating sorption type dehumidifiers and requested the Committee on Research to allocate funds for carrying out field tests of a solid and a liquid sorbent dehumidifier to determine if the proposed method was practicable and workable.

Meetings

The Director of Research, together with C. M. Humphreys and G. V. Parmelee, presented a paper entitled *The Role of Weather Data in Heating and Air Conditioning*, at the Conference on Weather and the Building Industry, arranged by the Building Research Advisory Board (BRAB) and held in Washington, D. C., January 12, 1950.

During the year the Director of Research spoke before the Statistical Exchange of Southern Florida at Miami; the Air Conditioning Conference at the University of Florida, Gainesville; the Warm Air Conference at Iowa State College; and the *Institute of Boiler and Radiator Manufacturers* at Absecon, N. J. He also represented the Society at the U. S. Technical Conference on Air Pollution in Washington in May; at a conference at the Library of Congress in November; and at the *A.S.R.E.* annual meeting in December. Staff members addressed Society chapters at Philadelphia, Dayton, North Carolina, and Indianapolis.

Acknowledgments

The Research Laboratory has been able to make substantial progress in 1950 in a number of important projects. Some projects have been brought to a successful conclusion. This has been due in no small measure to the encouragement and support given to the staff by the President and the members of the Council, and especially by the chairman and members of the Committee on Research and the members of the various Technical Advisory Committees. The Director wishes to express the sincere appreciation of the 25 members of the research staff for this support and to express his thanks to all staff members for their loyal service and cooperation.

The session was concluded with a presentation of two technical papers, one by the author, and one by title.

Third Session—Tuesday, January 23, 2:30 P. M.

Second Vice President Ernest Szekely presided at the third session at which three technical papers were presented and discussed.

Fourth Session—Wednesday, January 24, 9:30 A. M.

Pres. Lester T. Avery opened the symposium on *Man And His Relation To Air* and outlined the work which the Society has been doing both in its laboratory and in cooperating institutions on physiologolical research, and the part the technical advisory committees have played in these endeavors. The symposium was moderated by Prof. Nathaniel Glickman, Chicago, Ill. who

introduced the speakers for a discussion of different aspects of the subject. Eugene F. DuBois, M.D., New York, spoke on *Physiological Aspects of Heating And Ventilating.*

He stated that it is difficult to establish standards for comfort that cover more than one narrow set of conditions. The factors to be considered are many. Concerning the human subject:—age, sex, clothing, previous activity, present activity, time of exposure, nationality, habit, acclimatization, etc. Concerning the room:—dry bulb temperature, humidity, air movement, radiation, etc.

The comfort zone is a relatively narrow band of temperature conditions where the heat loss from the body is most easily regulated to equal the heat production. There is little or no change in the temperature of the body tissue, no sweating, and the skin is about 91.5 F (33 C). At colder environmental temperatures the blood flow in the periphery is restricted and the skin is transformed from an active heat interchanger into a suit of leather clothing. Since one half of the body mass lies within one inch of the surface there is a large reservoir from which heat can be lost before the temperature of the core of the body is much affected.

Above the comfort zone, as the air temperature gets higher and higher the blood flow in the skin increases rapidly. Sweating increases and finally the only possible channel of heat loss is vaporization.

If a person who has been in a warm room with a high rate of vaporization goes into a cold environment there is a condensation of moisture in the cold layer of clothing. Some of this heat is returned to the body for a short time but later the rate of cooling is increased. When comparing two different environments the time element is important.

Charles S. Leopold, Philadelphia, Pa., presented a discussion on *The Engineering Aspects of Comfort Data.*

Mr. Leopold discussed the work of the Society in establishing optimum summer and winter conditions for a group by subjective testing, and outlined aspects of the subjective evaluation of comfort. He compared comfort for an individual with the statistical optimum for a group. His conclusions that the optimum conditions indicated in the Society work, of 75 F to 77 F in summer; 73 F to 75 F in winter, when the relative humidity is approximately 50 percent and 30 percent respectively, are adequately substantiated by field experience.

He pointed out the difference between the importance of variables in their effect on comfort when measured as departures from the optimum or from some condition which is too warm or too cool.

The importance of horizontal symmetry in radiation, particularly the local effect of hot or cold walls was explained.

The importance of the absence of a rational relation between desirable indoor and outdoor temperatures from hour to hour was also included as well as intolerance for perceptible air motion.

In conclusion he demonstrated the importance of incentive and time within the conditioned space in evaluating the effects of air conditioning.

Theodore F. Hatch, Professor of Industrial Health Engineering, University of Pittsburgh, Pittsburgh, Pa., discussed the *Standards For Industrial Heat Control,* and said that—

The limits of the thermal comfort zone are well established and, similarly, the upper limits of tolerance to acute heat exposures have been fixed, both in the laboratory and in industrial experience. Exposures in the traditional hot industries are now kept generally below this tolerance limit. The ill-effects of heat are so widely appreciated that, even in those industries having no unusual heat production, comfort-conditioning is becoming common.

Between the two limits of comfort and maximum physiological tolerance, however, there exists a considerable region within which many industries must operate—an area which is not well charted in respect to standards for the measurement of heat effects.

Beyond a general desire to reduce heat exposures as much as possible, engineers do not find in this region an adequate, quantitative basis for estimating the benefits which are to be derived from further heat control in comparison with the cost of such improvements. The elementary physiological criteria which serve for measuring acute effects are of limited use in the region of non-acute response. Similarly, the subjective criteria of comfort are of little help when there is no practical possibility of reducing the thermal environment to this degree. Responsibility for the health and safety of workers is met when the heat level is still high compared with the comfort limit. Below this level the problem is one of balancing the cost of control against the returns, measured in terms of increased efficiency and productivity and improved human relations. The potential benefits vary with the nature of the industry, increasing with the fineness of the work, the manual skills required and with the degree of concentration and mental effort involved.

The area between the comfort zone and the physiological tolerance limit can be divided into three overlapping zones wherein the effects of primary concern may be classed successively as (1) physiological, (2) psychophysiological and (3) psychological. In industry, the three sub-areas are of corresponding primary interest to these departments: (1) medical, (2) engineering and (3) management (industrial relations). The need exists for thermal physiologists and experimental psychologists to orient their work more toward industry, to provide a quantitative basis for defining and evaulating the varying heat effects and for establishing useful thermal standards between the upper limit of physiological tolerance and the lower limit of simple comfort, where no problem exists.

Robert W. Keeton, M.D., Chicago, Illinois, elected to speak on *Air Conditioning in the Treatment and Prevention of Disease.*

He stated that individuals singly and in groups indulge in coughing and sneezing through certain months of the year. Respiratory infections are more prevalent at such times. The inference follows that the infectious agents are airborne and the incidence of disease could be reduced by sterilization of the air. This can be accomplished by ultra-violet radiation and by atomizing triethylene glycol into an atmosphere with a 50 percent relative humidity. Such measures have not significantly reduced the incidence of respiratory disease. Respiratory infections can be spread directly by contact and indirectly by dust which has been formed from droplets coughed onto clothes, bed linens, floors and furniture. A space conditioned for comfort with a 50 percent to 60 percent relative humidity, free of drafts and unusual sources of cold radiation, furnishes a uniform environment. This does not cause physiologic stress and it does not produce nasal congestion, which may be a factor in initiating head colds.

Anesthesia tends to alter the heat regulating mechanism of the body. In the summer time, when the humidity is high, post-operative patients commonly develop heat retention which, if unrelieved, leads to heat strokes. This causes adrenal stimulation and, if long continued, presumably adrenal exhaustion. A residence of 48 hours in the air-conditioned recovery ward avoids such complications. In a study of convalescence it was noted that postoperative patients maintained in comfortable environment made unusually good recoveries. Patients producing extra heat as a result of hyperthyroidism may slip into an hyperthyroid crisis. The latter is prevented or ameliorated by exposure to a cool dry air.

In the waves of hot humid weather patients with heart disease are quite prone to die. The primary difficulty seems to be inability to transport heat to the surface due to circulatory impairment. Heat retention and death follow. This can be prevented by exposure to comfortable or cool dry environments.

In patients with coronary sclerosis, dehydration as the result of excessive sweating may precipitate a blood clot (coronary thrombosis) at the site of an arteriosclerotic plaque.

More recently the reduction to 92 F of the rectal temperature of children with congenital heart disease, who have a limited capacity for oxygenating their blood, has made it possible to operate some seemingly inoperable cases.

Air-conditioned environments are necessary for research studies which include work experiments throughout the year, alterations in circulation, and in the partition and distribution of electrolytes.

Captain Oscar Schneider (MC USN), Washington, D. C. discussed the *Problems Following Atomic Blasts.*

He declared that to provide a simplified picture of a broad and complex subject, it is best to regard the casualty-producing effects of an atomic explosion as being largely dependent on its ability to produce mechanical destruction. Thus, in the event of an air burst such as those employed in Japan, it can be expected that most of the fatalities will occur among persons who are injured by the collapse of buildings and houses, and by being unable to escape from the fires which sweep such ruined structures. In a ground surface burst, the area of blast damage will be less than in an air burst, and the number of casualties will therefore be less. In the event of an underwater burst, the damage from blast, as well as from fires will be still less, and so will the number of casualties.

Radioactivity from an atomic explosion is serious, but not by any means serious enough to justify all of the worry it has occasioned. After a blast which has come with no warning, only about 15 percent of those sustaining fatal injuries will die from radiation illness. The remaining 85 percent will be victims of trauma and burns. There is virtually no lingering radioactivity from an air burst. After the first 90 seconds, the radiation effects from this type of explosion are no longer a source of danger. Owing to the deposit of fission products and unfissioned atoms of bomb material on the ground or in the water, radiation injuries may form a greater percentage of casualties from a ground or underwater burst, but this percentage can be reduced by an effective Civil Defense Organization. Properly designed ventilation systems in buildings and shelters can also reduce this radioactivity hazard.

The effects of radiation are essentially those due to internal hemorrhage and anemia, complicated by infection. Proper medical care can reduce the mortality rate among the victims with borderline amounts of irradiation as well as those with other serious injuries. Accordingly, the perfection of arrangements for the rescue, first aid, and hospitalization of the injured is of paramount importance in preparation for an atomic war. There is no convincing evidence that atomic radiation will increase the cancer incidence among its victims or even that any noticeable hereditary effects will occur. In Japan the sterility which was observed in many of the injured proved, in all of the cases examined, to be only temporary.

In preparing for an atomic war, individuals can best provide protection for themselves, their families, and their communities by (1) cooperating in every way with their Civil Defense Organizations, (2) learning the simple rules of taking shelter, of first aid, and of accident-prevention in the home, and (3) schooling themselves to keep calm in an emergency and to avoid creating a panic.

Following the talks there was discussion from the floor by many members. The panel answered the questions presented by the audience.

President Avery thanked the participants in the discussion, both the symposium members and those from the floor.

FIFTH SESSION—WEDNESDAY, JANUARY 24, 2:30 P. M.

At the fifth session, presided over by First Vice President L. E. Seeley, three technical papers were presented by their authors, and one paper was presented by title only.

SIXTH SESSION—THURSDAY, JANUARY 25, 9:30 A. M.

At the final session of the meeting, Treas. Reg F. Taylor presided and three technical papers were presented.

President Avery then conducted the installation of the new officers and Council members: The new officers—Howard E. Sproull, treasurer; Reg F. Taylor, 2nd vice president; Ernest Szekeley, 1st vice president; and Lauren E. Seeley,

president; council members: John H. Fox, L. N. Hunter; B. H. Spurlock, Jr., and Neil H. Peterson.

President Avery gave the gavel to President-elect Seeley which the latter accepted by saying that he was looking forward to serving the Society during the coming year, and hoped to justify the members' confidence in electing him to the office. President Seeley called on S. W. Beaty, Chairman of the Resolutions Committee, who presented the following resolutions which he moved for adoption. The motion, seconded by Past Pres. Lester T. Avery, was adopted unanimously.

Resolutions

Whereas: The 57th Annual Meeting of the A.S.H.V.E., January 22-25, 1951, has attained new highs in accomplishments both technically and socially, therefore, Be It Resolved That We:

1. Wholeheartedly recognize and express our appreciation for the cooperation of the National Officers and Staff, the Philadelphia Chapter and the efficient committees of the Committee on Arrangements which actuated careful planning into accomplished fact insuring the profit and pleasure which have highlighted this meeting.

2. Thank particularly the many members of the Philadelphia Chapter whose various personnel assisted in registration, meetings and the Exposition, the Philadelphia Convention and Visitors Bureau for its assistance in providing the housing bureau and the Hotels whose management cooperated in furnishing such excellent service and facilities for social functions.

3. Thank the authors of papers and speakers who have devoted so much time to their preparation of data and their informative presentations of new progress in the arts of heating, ventilating and air conditioning, and also commend Dr. Lillian Gilbreth and Cameron Ralston for their inspiring messages.

4. Also express appreciation to the daily newspapers and the technical press for their part in reporting the meeting to the public and the profession; and compliment the manufacturers who exhibited at the Exposition in the Commercial Museum for their excellent displays, and the managers and their staff.

5. Express particular gratitude for the gracious entertainment thoroughly enjoyed by our visiting ladies which brought to us the happy discovery that the slogan, "The City of Brotherly Love" expresses only half of it and should be amended to read "The City of Brotherly and Sisterly Love."

Respectfully Submitted,

The Resolutions Committee
S. W. Beaty, *Chairman*, Shreveport, La.
I. W. Cotton, Indianapolis, Ind.
R. C. Chewning, Portland, Ore.

The 57th Annual Meeting of the Society was declared adjourned at 11:15 A. M.

PROGRAM—57th ANNUAL MEETING

The Bellevue-Stratford—The Benjamin Franklin

Warwick Hotel—Convention Hall

Philadelphia, Pa., January 22-25, 1951

Saturday—January 20

9:30 a.m. Charter and By-Laws Committee, H. E. Sproull, *Chairman* (Bellevue-Stratford—*Room 108*)

10:00 a.m. Committee on Research, R. C. Cross, *Chairman* (Bellevue-Stratford—*Green Room*)

1:30 p.m. Executive Committee of Council (Bellevue-Stratford—*Room 108*)

8:00 p.m. Ways and Means Committee Meeting, Ernest Szekely, *Chairman* (Bellevue-Stratford—*Blue Room*)

Sunday—January 21

9:00 a.m. Admission and Advancement Committee Meeting (Bellevue-Stratford—*Room 104*)

10:00 a.m. REGISTRATION (Bellevue-Stratford—*Oak Room*)

10:00 a.m. Council Meeting (Bellevue-Stratford—*Green Room*)

10:00 a.m. TAC on Air Cleaning, R. S. Farr, *Chairman* (Bellevue-Stratford—*Room 105*)

3:30 p.m. Informal Get-Together of Chapters' Conference Committee Members, B. L. Evans, *Chairman* (Bellevue-Stratford—*Pink Room*)

4:00 p.m. Committee on Research, R. C. Cross, *Chairman* (Bellevue-Stratford—*Room 108*)

4:00 p.m. Welcome Tea (Bellevue-Stratford—*Reception Room,* 18th Floor)

Monday—January 22

9:00 a.m. REGISTRATION (Bellevue-Stratford—*Oak Room*)

9:00 a.m. Committee on Estimating Fuel Requirements (Bellevue-Stratford—*Room 106*)

9:30 a.m. FIRST SESSION (Bellevue-Stratford—*Rose Garden*) Pres. Lester T. Avery presiding

- Welcome to Philadelphia
- Response by President Avery
- Reports of Officers and Council
- Losses from a Floor-Type Panel Heating System, by F. W. Hutchinson, Berkeley, Calif., D. L. Mills and L. J. La Tart, Rome, N. Y., presented by Mr. Mills
- Operating Experience and Data from Sidewalk Snow-Melting System, by L. A. Stevens and G. D. Winans, Detroit, Mich., presented by Mr. Winans
- Design Factors in Panel and Air Cooling Systems, by C. S. Leopold, Philadelphia, Pa., presented by the author

12:15 p.m. GET-TOGETHER LUNCHEON (Bellevue-Stratford—*Ballroom*)

Speaker: Dr. Lillian M. Gilbreth, Consulting Engineer

Subject: "Management Problems in the Home"

2:00 p.m. EXPOSITION—Opening of 10th International Heating and Ventilating Exposition (Commercial Museum, 34th below Spruce St.)
Exhibit will open daily from 12 noon to 10:00 p.m., Tuesday to Thursday, and will close at 6:00 p.m. on Friday
A.S.H.V.E. Booth No. 88

2:00 p.m. Chapters' Conference Committee, B. L. Evans, *Chairman* (Bellevue-Stratford—*Junior Room*)

2:00 p.m. TAC on Air Distribution, Ernest Szekely, *Chairman* (Bellevue-Stratford—*Room 104*)

2:00 p.m. TAC on Combustion, T. H. Smoot, *Chairman* (Bellevue-Stratford—*Room 108*)

2:00 p.m. TAC on Cooling Load, W. E. Zieber, *Chairman* (Bellevue-Stratford—*Room 105*)

2:00 p.m. TAC on Panel Heating and Cooling, P. B. Gordon, *Chairman; Group A*, A. B. Algren, *Chairman* (Bellevue-Stratford—*Green Room*)

3:00 p.m. Committee on Testing and Rating Heavy Duty Furnaces, E. K. Campbell, *Chairman* (Bellevue-Stratford—*Room 107*)

3:00 p.m. Ladies' Show and Tea at John Wanamaker's (*Greek Room*)

4:00 p.m. TAC on Weather Design Conditions, W. M. Wallace, II, *Chairman* (Bellevue-Stratford—*Blue Room*)

6:30 p.m. "A Night with the Pennsylvania Dutch" (Benjamin Franklin—*Ballroom*)

7:00 p.m. TAC on Combustion, T. H. Smoot, *Chairman* (Bellevue-Stratford—*Room 104*)

Tuesday—January 23

8:30 a.m. TAC on the Heat Pump, R. C. Jordan, *Chairman* (Bellevue-Stratford—*Room 107*)

9:00 a.m. REGISTRATION (Bellevue-Stratford—*Oak Room*)

9:30 a.m. TECHNICAL SESSION (Bellevue-Stratford—*Rose Garden*) First Vice Pres. Lauren E. Seeley presiding
Report of Tellers of Election
The Efficiency of Bituminous Coal-Burning Space Heaters, by J. W. Tieman and F. L. Bagby, Columbus, Ohio, presented by Mr. Bagby
The Measurement of Smoke, by K. O. Beatty, Jr. and J. E. Deas, Jr., Raleigh, N. C., presented by Prof. Beatty
Report of Committee on Research, R. C. Cross, *Chairman*
Rating of Fuel Oils by a Test Unit, by D. W. Locklin and G. V. Parmelee, Cleveland, Ohio, presented by Mr. Locklin
Smoke Measurement in a Fuel Oil Test Unit, by D. W. Locklin and G. V. Parmelee, presented by title

9:30 a.m. TAC on Industrial Ventilation, W. N. Witheridge, *Chairman* (Bellevue-Stratford—*Room 108*)

9:30 a.m. TAC on Sensations of Comfort, C. S. Leopold, *Chairman* (Bellevue-Stratford—*Room 106*)

9:30 a.m. TAC on Sound Control, T. A. Walters, *Chairman* (Bellevue-Stratford—*Room 105*)

10:30 a.m. Ladies "Brunch" (Warwick Hotel—*Ballroom*)

11:30 a.m. Bus Tour of "Old and New Philadelphia and Historic Valley Forge"

12:15 p.m. Luncheon Meeting—TAC on Hot Water and Steam Heating, J. W. James, *Chairman* (Bellevue-Stratford—*Red Room*)

2:00 p.m. TECHNICAL SESSION (Convention Hall—*Ballroom*) Second Vice Pres. Ernest Szekely presiding

A Survey of Electrostatic Precipitation, by E. A. Walker and J. E. Coolidge, State College, Pa., presented by Professor Walker

Theory of Earth Heat Exchangers for the Heat Pump, by L. R. Ingersoll, F. T. Adler, A. C. Ingersoll, Madison, Wis., and H. J. Plass, Stanford University, Calif., presented by Prof. L. R. Ingersoll

Factors Useful in Ground Grid Design for Heat Pumps, by George S. Smith, Seattle, Wash., presented by the author

2:30 p.m. TAC on Heating Load, A. I. Brown, *Chairman* (Convention Hall—*Room 303*)

2:30 p.m. TAC on Sorbents, John Everetts, Jr., *Chairman* (Convention Hall—*Room 307*)

4:30 p.m. Committee on Research (Convention Hall—*Room 305*)

7:00 p.m. Past Presidents' Dinner (The Union League of Philadelphia, Broad & Sansom Sts.)

7:30 p.m. Chapters' Conference Committee, B. L. Evans, *Chairman* (Bellevue-Stratford—*Junior Room*)

7:30 p.m. Guide Committee Meeting (Bellevue-Stratford—*Room 107*)

Wednesday—January 24

8:00 a.m. TAC on Odors, T. H. Urdahl, *Chairman* (Bellevue-Stratford—*Room 108*)

8:00 a.m. Breakfast Meeting—TAC on Panel Heating and Cooling—*Group D*, J. S. Locke, *Chairman* (Bellevue-Stratford—*Room 107*)

9:00 a.m. REGISTRATION (Bellevue-Stratford—*Oak Room*)

9:00 a.m. TAC on Insulation, M. W. Keyes, *Chairman* (Bellevue-Stratford—*Junior Room*)

9:30 a.m. TECHNICAL SESSION (Bellevue-Stratford—*Rose Garden*) Pres. Lester T. Avery presiding

SYMPOSIUM—Nathaniel Glickman, Moderator

Subject: "Man and His Relationship to Air"

1. EUGENE F. DUBOIS, M.D., New York—The Physiological Aspects of Heating and Ventilating.

2. CHARLES S. LEOPOLD, Consulting Engineer, Philadelphia Pa.—The Engineering Aspects of Comfort Data.

3. THEODORE F. HATCH, Professor of Industrial Health Engineering at University of Pittsburgh, Pittsburgh, Pa.—Standards for Industrial Heat Control.

4. ROBERT W. KEETON, M.D., Chicago, Ill.—Air Conditioning in the Treatment and Prevention of Disease.

5. CAPT. OSCAR SCHNEIDER, (MC) USN, Washington, D. C.—The Problems Following Atomic Blasts.

9:30 a.m. TAC on Heat Flow through Glass, R. A. Miller, *Chairman* (Bellevue-Stratford—*Room 104*)

12:15 p.m. Luncheon Meeting—Nominating Committee (Bellevue-Stratford—*Blue Room*)

12:30 p.m. Luncheon and Fashion Show (Warwick Hotel—*Ballroom*)

2:00 p.m. TECHNICAL SESSION (Convention Hall—*Ballroom*) First Vice Pres. Lauren E. Seeley presiding

Field Studies of Heat Losses from Concrete Floor Panels, by C. M.

Humphreys, C. V. Franks and L. F. Schutrum, Cleveland, Ohio, presented by Mr. Schutrum

Principles of Effective Heat for Steam Heated Structures, by C. O. Mackey, N. R. Gay, Ithaca, N. Y., R. D. Tutt, New Britain, Conn., E. G. Powell, Chicago, Ill., and E. L. Broderick, Mt. Vernon, N. Y., presented by Professor Mackey

Transportation Heating and Air Conditioning Systems and Controls, by N. O. Kirkby, Chicago, Ill., presented by the author

Water-Vapor Permeability of Building Papers and Other Sheet Materials, by E. R. Bell, M. G. Seidl and N. T. Krueger, Madison, Wis., presented by title

7:00 p.m. ANNUAL BANQUET (Bellevue-Stratford—*Ballroom*)
Toastmaster: Merrill F. Blankin, past president of A.S.H.V.E.
Speaker: Cameron Ralston, Utica, N. Y.
Subject: "The Plagues of Democracy"
Presentation of Past President's Emblem to Lester T. Avery
Announcement of Award of F. Paul Anderson Medal by Lester T. Avery

Thursday—January 25

9:00 a.m. REGISTRATION (Bellevue-Stratford—*Oak Room*)

9:30 a.m. TECHNICAL SESSION (Bellevue-Stratford—*Rose Garden*) Treas. Reg F. Taylor presiding

Use of Aircraft Propellers for Axial Flow Fans, by E. C. Lundquist, Iowa City, Iowa, and M. J. Hamilton, Santa Monica, Calif., presented by Professor Lundquist

Roof Spray for Reduction in Transmitted Solar Radiation, by G. E. Sutton, Gainesville, Fla., presented by the author

Solar Energy Transmittance of Figured Rolled Glass, by G. V. Parmelee and W. W. Aubele, Cleveland, Ohio, presented by Mr. Parmelee

Installation of Officers

Unfinished Business

New Business

Resolutions

Adjournment

10:30 a.m. Ladies Bus Trip to Atlantic City

COMMITTEE ON ARRANGEMENTS

A. J. NESBITT, *General Chairman*
M. F. BLANKIN, *Honorary Chairman*
F. H. BUZZARD, *Vice Chairman*

Banquet—M. E. Barnard, *Chairman;* H. R. James, E. C. Lambert, C. J. Lubking, W. W. Malloy, H. H. Mather, G. W. Powell, Jr., G. E. Tuckerman, F. B. Wilder.

Entertainment—E. K. Wagner, *Chairman;* R. H. Ackermann, R. J. Augusterfer, H. H. Erickson, H. J. Faltenbacher, H. W. Goldner, J. H. Hucker, R. F. Hunger, L. P. Hynes, M. G. Kershaw, R. S. Mizener, S. R. Nusbaum, F. F. Pfeiffer, A. M. Robertson, Karl Rugart, Edward Semel, R. J. Sigel, J. K. Walters.

Exposition—C. F. Dietz, *Chairman;* C. G. Binder, H. W. Doerrfuss, C. J. Forve, Nicola Ginzburg, A. A. Granke, H. E.

Grossman, J. E. Heim, J. J. Hucker, Jr., H. K. King, A. E. Kriebel, H. H. Mohrfeld, A. G. Mutimer, H. M. Parent, R. M. Sharp, G. F. Smith, W. D. Strause, H. N. Teuber.

Finance—A. C. Caldwell, *Chairman;* C. B. Eastman, Edwin Elliot, R. J. Lee, L. E. Moody.

Ladies—E. H. Dafter, *Chairman;* G. F. Bertrand, R. H. Smiles. *Assisted by:*—Mmes. R. S. L. Arnold, M. E. Barnard, G. F. Bertrand, M. F. Blankin, F. H. Buzzard, L. M. Church, W. P. Culbert, E. H. Dafter, C. F. Dietz, John Everetts, Jr., H. B. Hedges, J. H. Hucker, R. F. Hunger, M. G. Kershaw, J. O. Kirkbride, C. S. Leopold, Ludwig Mack, H. H. Mather, J. W. McElgin, L. E. Moody, A. J. Nesbitt, W. S. Scott, E. K. Wagner.

Publicity—W. P. Culbert, *Chairman;* G. P. Alexander, A. E. D'Ambly, S. E. Plewes, R. D. Touton.

Reception—J. O. Kirkbride, *Chairman;* L. M. Church, *Co-Chairman;* R. S. L. Arnold, T. J. Atkins, Lucien Buck, P. A. Cavanagh, S. E. Farrington, H. E. Gross, H. B. Hedges, A. H. Koch, James Marshall, H. G. McCullough, F. B. Millham, H. B. Prewitt, J. J. Quinn, R. P. Schoenijahn, W. S. Scott, C. D. Shields, Pierce Timmis, L. A. Tucker.

Sessions—John Everetts, Jr., *Chairman;* Alan Dombro, R. E. Sexton, M. A. Sheffler.

Transportation — Isadore Director, *Chairman;* E. I. Curley, F. N. Darby, L. C. Davidson, J. E. Hutchison, Ludwig Mack, P. H. Yeomans.

1407

No. 1407

LOSSES FROM A FLOOR-TYPE PANEL HEATING SYSTEM

By F. W. HUTCHINSON*, BERKELEY, CALIF., D. L. MILLS**, AND L. J. LA TART†, ROME, N. Y.

TEN YEARS ago a research program was initiated at the University of California to investigate analytically and experimentally the effectiveness and operating characteristics of various types of panel heating and cooling systems. The part of this program which dealt with fundamentals of design was carried on cooperatively under the sponsorship of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; subsequent projects have been undertaken with respect to the rating and thermal characteristics of various types of panel construction.

As the program developed it became evident that wide discrepancy existed between the results obtained in laboratory tests and the results which were said to occur in actual installations. Direct comparison was difficult, however, because precise experimental methods can only be used on a field installation when adequate and costly instrumentation is provided. Three years ago an exceptional opportunity developed for carrying out an extensive experimental check on an actual installation, but with instrumentation sufficient to permit achieving laboratory accuracy. At that time a new building 25 ft by 60 ft (with an 11 ft 6 in. ceiling) was made available. Although this structure was intended for regular occupancy it was nonetheless equipped at the time of erection with some hundreds of embedded thermocouples and with all other instruments and controls needed for accurate testing of the heating system. Detailed descriptions of the building[1] and of its heating system[2, 3] and instrumentation[4] are available in the literature. Various technical articles from this *field laboratory* have presented results of studies on panel rating[5] and on the influence of embedded depth[6] on uniformity of surface temperature.

As tests of the actual system proceeded it became evident that losses of heat either through the floor slab or around the periphery were very much greater than indicated by common belief. During the first heating season that the

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** Research and Development Engineer, Revere Copper & Brass, Inc. Member of A.S.H.V.E.

† Research Department, Revere Copper & Brass, Inc.

[1] Exponent numerals refer to Bibliography.

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

system was in operation losses were estimated at from 25 to 35 percent, but data from these early tests were not considered sufficiently conclusive to justify publication. During the 1948-1949 and the 1949-1950 heating seasons, the subject of losses was selected for comprehensive investigation. This paper presents the results of the study. In addition to the loss data, experimental results are also presented giving the actual rating of unit area of floor panel and the combined film coefficient of heat transfer for such a panel as evaluated from tests on the actual system; the latter results are of particular interest in that they permit conclusive decision as to the applicability of the laboratory tests of other investigators to actual field installations.

SYSTEM THERMAL CHARACTERISTICS

The heating system on which a majority of the tests were run consisted of a sinuous coil floor panel occupying 118 sq ft of surface in a room having 220 sq ft of floor area. The coil was fabricated of ¾ in. copper tube (⅞ in. O.D.) on 9 in. centers, and was embedded in a 4 in. concrete slab poured on top of a 6 in. fill of gravel. The coil was leveled so that the distance from the top of the tube to the panel surface was ½ in. As an overall check on the results of tests from this one system a few additional tests were made on a different panel of equal area and the same construction, but with an embedded depth of 1 in. rather than ½ in.

Data from the two systems were recorded throughout the two heating seasons, but for the purposes of loss determination only a very few periods were suitable for complete tests. In order for data on losses to have general significance it is essential that conditions closely approaching equilibrium exist at the time of the test. With a heated floor slab, however, equilibrium is rarely realized as each change in weather necessitates altering the energy input to the slab and each change in slab temperature leads to readjustment of temperature gradients throughout a huge mass of earth extending some feet below the gravel fill on which the slab rests. The earth has a relatively low thermal conductivity and a relatively high thermal capacity (product of specific heat and of density) so long time lags intervene between a thermal change and re-establishment of thermal equilibrium. Even when weather changes are of a minor degree there may be a significant adjustment of gradients immediately below the slab with consequent departure of instantaneous loss rates from the more significant average values.

During periods in which the heating load is increasing, losses will occur at rates higher than those corresponding to equilibrium, whereas during intervals of decreasing load the rate of losses will be much less than equilibrium values. Considered for the heating season as a whole these transient losses seem to average at a value not greatly different from the integrated seasonal value based on assumed maintenance of steady state, but use of the instantaneous transient rates would lead to serious error. As an example of the importance of this circumstance consider a floor slab which is operating at steady load with a surface temperature of 80 F; the earth immediately below the slab will be at some equilibrium temperature (the value depending on type and thickness of slab as well as on the type of fill and the type of soil) usually somewhat higher than the panel surface temperature. If, now, the load on the system drops sharply, the heat supply to the panel would be cut off and as equilibrium tended

to be restored the rate of instantaneous loss to the earth would decrease and in some cases, would reverse.

During test runs on the actual system under conditions of decreasing load, negative loss rates were repeatedly observed, thus showing that for such transient periods heat was flowing from storage in the earth back up through the slab. In such cases the energy which at some earlier period was *lost* from the slab is now regained and effectively utilized in meeting part of the heating load. Unfortunately the reverse-flow usually occurs at a time when the actual heating load is very small; in such cases the returned *losses* may enter the heated space at a rate exceeding the heating requirements and thereby lead either

TABLE 1—EXPERIMENTAL DATA FROM ALL TESTS

TEST NUMBER		SYSTEM NO. 1			SYSTEM NO. 2
		No. 1	No. 2	No. 3	No. 1
Room air temperature at 48 in. level, t_a	F	67.0	68.5	70.0	74.0
Water temperature, entering coil, t_e	F	99.5	99.0	103.0	103.0
Water temperature, leaving coil, t_l	F	92.0	92.0	95.0	96.5
Panel surface temperature, t_{s1}	F	81.0	80.5	85.0	87.0
Panel surface temperature, t_{s2}	F	81.0	80.0	77.0	83.5
Panel surface temperature, t_{s3}	F	83.0	77.0	80.5	86.5
Panel surface temperature, t_{s4}	F	76.5	84.0	88.0	86.5
Panel surface temperature, t_{s5}	F	80.0	81.0	84.0	85.0
Panel surface temperature, t_{s6}	F	...	...	...	87.5
Flow rate to coil	gpm	1.16	1.16	1.16	1.20
Heat meter reading, Q_m	Btu/ sq ft/hr	26.9	25.1	29.4	23.4
Room floor area, A_f	sq ft	220	220	220	144
Unobstructed panel area, A_u	sq ft	110	110	110	106
Partially obstructed panel area, A_o	sq ft	0	0	0	5
Completely obstructed panel area, A_c	sq ft	8	8	8	7
Total panel area ($A_u + A_o + A_c$), A_p	sq ft	118	118	118	118

to overheating or to further losses due to the need of excess ventilation for maintaining a comfortably low room temperature.

For the above reasons, acceptable periods for testing slab losses are very few and much time and care must be devoted to selection of test periods during which the system is assuredly in a state reasonably close to that of equilibrium. Each of the tests reported in this paper was made near the end of a period of stable weather and at a time when thermal conditions within the room and within the slab were known (from periodic readings) to be practically constant.

PROCEDURE

Since the rooms under test were regularly occupied during the hours of the workday, the air temperature maintained was not established by the test group, but left under control of the occupants. As previously stated, tests were only made following periods of stable temperature, but over the two heating seasons the occupants did make many adjustments in room air temperature. Hence, the air temperatures shown for the various tests are not the same.

Table 1 presents a summary of all test data relevant to the present report. Three tests are reported for one heating system and a single check test for a

second heating system. In all cases air and surface temperatures were obtained from calibrated thermocouples whereas temperatures of the water entering and leaving the heating coil were obtained from calibrated thermometers (accurate to 0.1 F) and checked with thermocouple readings; both thermometers and thermocouples were installed in thermometer wells in the mains immediately ahead of and immediately following the heating coil. Flow was determined from readings of a calibrated meter located near the entrance to the coil. A standard surface-type heat meter, also calibrated, was used to obtain the total rate of heat loss (radiation plus convection) from unit area of the heated floor surface to the room. For the first system panel surface temperatures were recorded at five points (shown as t_{s1}, t_{s2}, t_{s3}, t_{s4}, and t_{s5} in Table 1); for the second system six such temperatures were recorded.

ANALYSIS OF TEST DATA

For purposes of the present report the word *loss* is used to designate the difference between the quantity of energy supplied to the heating panel and the quantity of energy transferred from the upper surface of the panel to the occupied space. (Refer to the section on *Discussion* for an explanation of the conditions for which this term may be a misnomer.) The total energy input to the panel is given by,

$$Q_t = (8.345)\ (60)\ (\text{gpm})\ (t_e - t_l) \quad \ldots\ldots\ldots\ (1)$$

where

Q_t = total heat input to the heating panel expressed in Btu per (square foot) (hour).
gpm = experimentally determined flow rate through the sinuous coil.
t_e = temperature of the water at entrance to the coil.
t_l = temperature of the water at exit from the coil.

The total heat transferred from the panel surface to the room in Btu per hour is given by the expression

$$Q_m A_p \quad \ldots\ldots\ldots\ (2)$$

where

Q_m = experimentally determined heat meter reading.
A_p = area of the panel.

Now let Z equal the percentage of total energy input which is lost from the edges and from the rear of the panel and note that $100 - Z$ is then equal to the percentage of input which is transferred from panel surface to the room; then

$$Q_m A_p / Q_t = (100 - Z)/100 \quad \ldots\ldots\ldots\ (3)$$

where

Z = the only unknown and hence can be solved for.

One difficulty exists, however, in the accurate evaluation of panel area. In the room tested, as would be true in most actual floor-panel heated rooms, some obstruction of panel surface existed. Of a total heated area of 118 sq ft, 8 sq ft (6.8 percent) was covered by a worktable with cabinets under it that extended to the floor. The heating effect realized from this completely obstructed area would obviously be something less than from other parts of the panel hence, if this reduction were not taken into account, the value of $Q_m A_p$ would be unduly

small and the resultant calculated value of the losses, Z, would be less than the true value. One method of correction would be to assume that the upward heat flow from the obstructed panel surface is some fraction of the upward flow from equal area of unobstructed surface. Considerable difficulty exists in determining a numerical value for this fraction, but if it is assumed to be conservatively small the resultant value of Z would be greater than the true value and one could then deduce that the true value of Z is necessarily between the values obtained, with uncorrected panel area and with corrected panel area. This procedure will be demonstrated with sample calculations for test No. 1 with the first heating system (data read from Table 1):

Substituting test data into Equation 1,

$$Q_t = (8.345)\ (60)\ (1.16)\ (99.5 - 92)$$
$$= 4350.$$

Substituting into Equation 3,

$$(26.9)\ (118)/(4350) = (100 - Z)/100 \text{ giving,}$$
$$Z = 27 \text{ percent for } \textit{uncorrected} \text{ panel area.}$$

To determine an expression for use with the corrected panel area, let X equal the fraction of edge and rear losses from unobstructed heating surface to upward transfer from unobstructed surface. Then the total energy input to unit area of unobstructed panel would be,

$$Q_m(1 + X) \quad \ldots \quad (4)$$

Now assume that losses from obstructed surface are the same as losses from unobstructed surface, but that upward transfer from such surface is only 10 percent of that from unobstructed surface; the total energy input to unit area of obstructed surface would then be,

$$Q_m(0.1 + X) \quad \ldots \quad (5)$$

and the total energy input to the entire heating panel would be,

$$Q_t = Q_m A_u(1 + X) + Q_m A_e(0.1 + X) \quad \ldots \quad (6)$$

where

A_u = area of unobstructed heating panel.
A_e = area of completely obstructed heating panel.

The only unknown in Equation 6 is X, hence, this term can be solved for and the losses, Z, subsequently calculated from the relationship,

$$Z = (100)\ (X)/(1 + X) \quad \ldots \quad (7)$$

Now substituting the test data for test No. 1 into Equation 6,

$$4350 = (26.9)\ (110)\ (1 + X) + (26.9)\ (8)\ (0.1 + X)$$

giving,

$$X = 0.434.$$

and then from Equation 7,

$$Z = (100)\ (0.434)/(1 + 0.434)$$
$$= 30.1 \text{ percent losses for corrected panel area.}$$

The true losses from the panel are therefore between 27 percent and 30 percent with the higher figure probably being closer to the true value.

The above sample calculations are representative of those for the first three tests which are reported (Table 1) for the first system. For the second system the method of calculation for uncorrected panel area is no different from the method for the first system, but for the corrected area an additional term appears in Equation 6 to take account of partial obstruction over a 5 sq ft area of panel due to cabinets that come within 2 in. of the floor. Here, again, an assumption is necessary as to the influence of the partial obstruction; upward transfer of heat from this section has been taken as 20 percent of that from unobstructed panel; the revised form of Equation 6 is then,

$$Q_t = Q_m A_u(1 + X) + Q_m A_c(0.1 + X) + Q_m A_o(0.2 + X) \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where

A_o = partially obstructed panel area.

An additional calculation permits evaluation of the combined film coefficient for radiant and convective transfer from the panel surface to the room. If the

TABLE 2—SUMMARY OF RESULTS

TEST NUMBER	SYSTEM NO. 1				SYSTEM NO. 2	
	No. 1	No. 2	No. 3	No. 4	No. 1	No. 2
% Loss (Based on uncorrected panel area)	27.0	27.4	25.4	...	29.4	...
% Loss (Based on corrected panel area)	30.1	30.2	28.6	...	33.5	...
Film coefficient (Btu/hr/sq ft/F deg)	2.02	2.09	2.28	2.28	2.23	2.20

experimentally determined panel surface temperatures given in Table 1 are averaged and if the average is identified as t_s, the film coefficient h will then be obtained from the equation

$$Q_m = h(t_s - t_a)$$

or

$$h = Q_m/(t_s - t_a) \text{ Btu per (hr) (sq ft) (F deg)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

DISCUSSION OF RESULTS

Table 2 summarizes the results obtained from the data of Table 1 by the methods described in the preceding section. Losses based on uncorrected panel area are seen to range from 25.4 percent to 27.4 percent for the first system whereas the corresponding losses based on corrected area range from 28.6 percent to 30.2 percent. Differences of this order of magnitude are of no practical significance and, considering the difficulty of achieving steady state in a system of this type, the extent of the agreement of test results seems almost fortuitously good.

For the second system the losses are indicated as being slightly greater than for the first, but since only one test is reported this difference cannot be interpreted with any degree of assurance. The main purpose of the single run on the second system was to provide an overall check on the results from the first system and the authors suggest that the extent of agreement between results

from the two systems is surprisingly close. From theory one would expect a slight increase in losses from the second system since the tubes are embedded at a depth of 1 in. as compared with ½ in. depth for the first system; this difference in systems may explain the indicated difference in results, but such an explanation cannot be considered conclusive on the basis of a single test.

Interpretation of the term *loss* requires recognition of the three ways in which energy leaves the sides and the rear of a heating panel. For the two systems on which tests were conducted the heating panel did not occupy the full floor area of the room, hence part of the energy which was *lost* from the heating panel undoubtedly served to warm adjacent unheated floor area and thereby contributed to the ultimate heating of the room. In terms of system effectiveness this fraction obviously is not a true loss. The remaining energy which escapes from sides and rear of the heating panel is either lost to the earth directly below the unit or is lost peripherally to the surrounding earth. In seeking means of increasing the thermal effectiveness of a floor panel, a first step would be to separate the peripheral and vertical losses and thereby determine the extent to which use of peripheral insulation would reduce the total losses of the system; it is hoped that data related to such a separation can be obtained during the 1950-1951 heating season and presented in a subsequent paper.

The film coefficients resulting from these tests (see Table 2) are of particular interest in that they provide direct substantiation, from an actual field installation, of the results obtained in the laboratory by Wilkes and Peterson[7]. Much disagreement has existed as to the rating of floor-type heating panels, but the present results seem to indicate that ratings of actual panels do not appreciably differ from ratings obtained in the laboratory under controlled conditions. For a panel designed to operate at surface temperature of 85 F in a room with air temperature of 70 F, both laboratory and field tests now show that the total upward heat transfer (the panel rating) will be from 30 to 35 Btu per (hr) (sq ft). There have been many expressions of opinion to the effect that ratings much higher than this occur, but so far as the authors have been able to determine there have never been any published data, obtained under precise test conditions, to substantiate belief in a rating higher than 35 Btu per (hr) (sq ft) for an 85 F floor panel in a 70 F room.

During the heating season preceding the one in which the heat loss studies were started, data were obtained from the same two systems and used in calculations[5] of film coefficients based on *assumed* losses of 10 percent. In Table 2 test No. 4 on the first system and test No. 2 on the second system present recalculated values of the film coefficients based on revised losses equal to the values obtained from the other tests; it will be noted that on this basis the earlier tests give film coefficients almost identical with those obtained from the series of tests reported in this paper.

CONCLUSIONS

Over a two year period tests were conducted on two floor-type panel heating systems installed in regularly occupied rooms of an industrial building. Both systems used ¾-in. sinuous coil copper tube on 9 in. centers and in each case the coil was embedded in a 4 in. concrete slab poured over a 6 in. gravel fill. No insulation was provided either under or at the sides of the slab, or of the structure, and in each case the heated section of floor was less than total floor

area. Cellular insulation, ¾ in., separated heated from unheated floor sections.

Edge and rear losses (which could not, in these tests, be separated) were found to agree very closely on all tests and to be of the order of 30 percent of total energy supplied to the heating system. Based on these results it is evident that some type of insulation is required with floor-type panel heating if the operating cost is to be kept at a reasonably low value.

The film coefficients of heat transfer calculated from test data agreed very closely with the values obtained by other investigators and suggest that an acceptable design value of combined (radiation and convection) film coefficient for floor heating panels is 2.1 Btu per (hr) (sq ft) (F deg temperature difference from panel surface to air).

ACKNOWLEDGMENT

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DISCUSSION

W. J. HAJEK, Pasadena, Calif. (WRITTEN): The facts so clearly brought out in this paper confirm the necessity of careful and complete analysis of all factors involved in the design of a floor panel heating system, before a correct decision can be made as to the best means of controlling the heated space.

It is obvious that insulation is necessary if greater efficiency is desired. It also appears that without such insulation, the matter of space temperature control appears impossible without considerable expensive means, during those periods when a reverse flow of energy takes place, that is, from earth to panel.

If the panel is insulated, control of the time factor by outdoor signals appears to offer the best approach to successful control. However, indoor anticipation may be satisfactory in many cases. Where the panel is *not* insulated, present methods of control will prove inadequate and perhaps bring headaches to those concerned.

Group D of the Technical Advisory Committee on Panel Heating and Cooling will no doubt make good use of these findings in their approach to the research work in controls.

L. B. NYE, JR., Washington, D. C. (WRITTEN): We were greatly interested in the report by Hutchinson, Mills, and LaTart on losses from a floor type panel heating system. Presented with the problem of estimating operating costs of similar systems, we made an operating cost study on sixty gas heated homes of this type three years ago.

These homes were in two different groups by two different builders. About one-half of them used a sinuous heating coil in a concrete slab and the remainder used a grid type coil similarly located. In each case, the slab was insulated at the edge with one inch of rigid insulating board.

Analyses of the monthly meter readings for the sixty houses over a twelve month period indicated the operating cost of the heating systems to average 33 percent higher than that of similar houses of equal heat loss which were heated by direct standing radiation or warm air systems.

The study also indicated that the cost per degree day was far higher in mild weather than in severe weather. Occupants reported that it was necessary to maintain a constant thermostat setting day and night because of the slow pickup afforded by the systems. From this we concluded that the excess operating cost was a result of high heat losses to the ground which made up an increasingly large percentage of the heat demand as the weather grew mild.

We have been waiting for some time for more precise tests to back up these observations and felt some justification in the conclusion of this paper that 30 percent of the total energy supplied to the heating system was lost to the ground.

I should hasten to add that in none of the homes studied in our test was there any complaint of unsatisfactory heating. Customers were uniformly pleased with the quality of heating received and inasmuch as the houses were small, the total operating cost was not a subject of complaint.

E. L. WEBER, Seattle, Wash. (WRITTEN): In our practical experience we always use an expansion joint around the outer perimeter of a floor slab panel installation. This joint also acts as insulation between the slab and outer wall. If the slab rests upon a footing, it is essential to place insulation between the slab and the top of footing.

In multi-story concrete buildings with monolithic concrete slabs and walls, we keep the coils back from 10 in. to 12 in. from the inside wall.

I consider a 1 in. coverage for floor coils too small. It should not be less than 1¾ in. especially if copper coils are used, which expand considerably more than the concrete and are liable to cause cracks in the floor. Slabs should be poured and finished monolithic in one operation. This avoids *heaving up* of the finishing slab.

B. LE B. MUSGRAVE, New York, N. Y. (WRITTEN): I would like to congratulate the authors on making a most valuable contribution to panel heating by producing actual results obtained in a field installation.

Many years ago when I received my training, experience and the crude testing methods of those days very quickly taught us to treat floor type panel heating on grade with the utmost respect and caution, for the very reasons which the authors have set forth here today. Even with 2-in. granulated cork slab insulation, which was cheap in pre-war days, and with 1-in. insulation between the edge of the slab and the outside wall and foundation, we always had to allow at least 20 percent loss downward. Experience has taught us that panel coils should be set with the top side 2-in. below the surface of the floor including the finish coat, but excluding finish when *non-insulating* materials such as terrazzo, tiles, asphalt etc. are to be the finish. With *insulating* materials such as wood block, cork tile, etc. the coils should be set with the top side 1-in. below the surface of the finish coat, but in spite of the coils being 1-in. nearer the surface, 5 percent reduction on emission has to be allowed for this latter method. I submit that the greater depth allows a more even temperature spread over the whole panel area by cutting down the temperature differential between

pipes, and the readings for system 2 in Table 1 where the coils were at a greater depth, seem to support our practice. I would like to know if the authors' tests support this reasoning.

Two years ago in connection with snow melting, I came across a well known commercial asphalt mix, which has ingredients incorporated which give it outstanding properties as a moisture barrier, and in addition giving better heat insulation than materials I had previously known. It is also being specified for floor panel heating systems with very great success. I mention this in case it might be of help to those who have not yet heard of it, since cork has become so expensive, and aluminum foil insulation may become difficult to obtain. Independent laboratory tests in June 1950 proved the superior qualities of this material. Asphalt as you know is warm to the touch, a good heat conductor, and yet easy on the feet—therefore excellently suited to floor panel heating.

The utmost discretion must be used when considering the use of floor panel heating for the reasons mentioned in the paper. It is not the answer to panel heating for all types of premises. It is suitable for factories, halls, churches, lobbies, etc., but not necessarily suitable in premises or homes where floor obstructions of the panel area which is not allowed for, may take place in the future. For example, built-in furniture may be installed tight to the floor over the panels, or heavy carpeting and underlays laid. The emission from the floor panel can be upset by this blanketing, and the feeling of comfort may be impaired. To counteract this, the floor panels have to be forced to a higher temperature with a subsequent increase in fuel consumption. One point put forward in favor of floor type panel heating is that it insures a warm floor. However, by proper designing with ceiling or wall panels, the floor is going to be warm anyway.

Please do not feel that, because of the precautions that have to be taken with floor panel heating, I am indirectly saying that it should never be used. There is a place for it, in the right type of premises. After all, because a certain type of lighting fitting gives excellent results in a factory, the electrical engineer does not use this exclusively for all premises.

J. M. VAN NIEUKERKEN, New York, N. Y.: Some of the comments that occurred to me when I read this paper have already been touched upon by Mr. Mills in his presentation. So apparently we see eye to eye, after all, on some of the points I wished to bring out in the discussion.

The authors have rendered the industry a real service by undertaking these tests and focusing attention on a subject about which there still is a good deal of uncertainty. The tests on their particular installation show that ground and end-losses may indeed reach values of 27 to as much as 33 percent, although they admit that a considerable portion of what they call the ground losses may actually be used to heat adjacent portions of the panel not equipped with coils.

In reading this report a number of questions occurred to me: (1) Why was only roughly 50 percent of the floor equipped with coils?

The direct losses to the ground underneath for a larger panel with a correspondingly reduced water temperature might have been the same due to the longer length of the coils, but the end losses, and therefore the total ground losses, would presumably have been reduced due to the lower temperature gradients.

The paper itself does not state what end insulation was provided. The magnitude of the ground losses, which in this case were understood to include the edge losses to adjoining panels, seems to indicate, however, that the insulation, consisting of 1 in. celotex strips separating the panels from one another and from the walls, was inadequate. The use of much wider strips of waterproofed edge insulation and the modern practice of providing such insulation for some two or three feet along the edges under the slabs would probably have yielded widely different values of the ground losses.

(2) Losses of 27 to 30 percent in the location of the test installation might have been much less elsewhere where ground conditions are more favorable. The authors do not state what their actual ground conditions are. The possible presence of moisture in the soil under the slab and the need under all circumstances of slab systems having thorough drainage should always be countenanced.

(3) No good designer would propose a ceiling type radiant heating installation without adequate insulation behind the coils, unless he belongs to those who want to straddle two horses by heating two stories simultaneously and thereby having no solid footing on either.

Why then should one wish to build floor slab installations without making sure that the slab is adequately insulated from the sub-soil below? This does not necessarily mean the need for extra insulation. In some localities the ground itself provides the insulation naturally, for instance, where the ground is highly porous and only thorough drainage is needed, perhaps by means of a shallow ditch around the building to divert surface water.

In certain areas in the east, where the ground conditions were found unusually favorable, may thousands of floor slab installations seem to have given a very satisfactory account of themselves. Complaints from other areas, however, show that in these localities the results were much less favorable, if not actually unsatisfactory.

Incidentally, the use of insulation under the concrete floor slab or under the wood floors over a crawl space, where ceiling panels provide the necessary heat, might in a number of cases have solved what experience has proved to be inadequate or merely skimpy design.

In some localities ground conditions are so severe that, in order to provide sufficient insulation for the surface slab, even a deep layer of broken stone or clean gravel is insufficient, and it may be necessary to top the latter by a sub-slab of, say, vermiculite concrete, or by several layers of asphalt paper made impervious by waterproofing compounds. On the other hand, in numerous instances the beds of broken stone or gravel, varying in thickness from 6 to 9 in. together with adequate water-proof edge insulation, have proved to provide all the insulation necessary for economical operation and comfort.

The authors have touched on an extremely important aspect of radiant heating; the industry therefore will look forward with great interest to their subsequent reports. Their studies should be of considerable help in eventually overcoming the uncertainties which greatly varying soil conditions in widely scattered geographical locations still impose on the floor slab type of radiant heating installations.

P. B. GORDON, New York, N. Y.: This paper is another confirmation of the need for some form of insulation behind any panel installation, whether it be a ceiling panel or a floor panel installation. This point has also been made by Mr. Van Nieukerken. Not only does this insulation affect the downward loss or reverse loss in this case, but it also affects the upward output into the room.

However, there are a few questions I should like to submit to the authors for discussion:

1. Is the outdoor temperature data available for the time of the test runs? The magnitude of the reverse losses is closely related to outdoor temperature.

2. Does the three-quarter inch insulation break mentioned in the paper that separates the heated floor section from the unheated floor section project below the slab? If not, there would be some effective heat flow from the edge of the heated panel into the adjacent panels thereby creating effective heating into the test room itself with resulting effect on values given for the surface conductance.

3. Is it not extremely difficult to obtain an average surface temperature for the panel as constructed? An embedded depth in the order of half an inch or one inch is much shallower than usual for panels with tubes on nine in. centers. Because of such wide spacing and very shallow embedded depth, the temperature spread between

the tubes is high, in the order of 4 or 5 deg, or more, and not readily determined by taking a few thermocouple readings, as indicated in the paper.

A slight discrepancy in this temperature difference would affect the heat output given in terms of Btu per (hr) (sq ft) per degree difference, as we are only talking of relatively small temperature differences for the combined convection and radiation release.

4. Was there any other heat release available in the test room during the period of the test run?

I am not too certain, though I believe when these rooms were originally built they also had ceiling panels. Any other form of heating in operation in the room during the period of test run, of course, would not only affect the heat release into the room but also affect the loss into the ground.

5. If there was no other heat release into the room, what is the comparison between the Btu per hr requirement for the space as calculated by THE GUIDE methods compared with the measured release from the panels?

6. I believe it to be a mistake to indicate heat outputs from floor panels or ceiling panels in terms of total Btu per (hr) (sq ft) per degree difference between air and surface temperature, or to use a combined film coefficient for radiation and convection release.

For example, with a 70 F room and with, say, a 63 F average unheated surface temperature for the other surfaces of the room, the total output might be in the order of 30 to 35 Btu per (hr) (sq ft). However, there are many rooms that have unheated mean radiant surface temperatures as high as 68 or 69 F, and likewise other rooms that have unheated average surface temperatures as low as 40 to 45 F. Therefore, the 15 deg temperature difference between room and air and panel surface is no longer a valid method of expressing the combined heat output.

7. With regard to reverse loss from the panel, and in this case into the ground, should we not divide the total reverse loss into two items: one, the flow of heat that would go into the ground because we have a 70 F room built on the ground? Any room built on the ground with any form of heating will have some loss into the ground. The remainder of the total loss would then result from the warm water piping system built on the ground. Otherwise, we are liable to distort our thinking by adding these two items which represent the 27 or 30 percent total reverse loss that is used in the paper. The question is—how would this loss compare with the flow into the ground for a panel ceiling system, an air heating system, an exposed radiator system, or a convective system?

8. Is there justification for deducting the obstructed area? I believe some of the heat flow up through the obstructed area eventually ended up in the room as heat release, or blocked the flow of heat to the outside of the obstructed area thereby affecting the loss calculation.

The authors are to be thanked for a valuable contribution to the fund of data on this subject that is now being presented for critical review.

W. P. CHAPMAN, Pittsburgh, Pa.: I have three comments to make: (1) the importance of $Q_m A_p$, (2) the evaluation of Q_m, and (3) the evaluation of A_p.

The back losses are based on the term $Q_m A_p$; therefore, the exact evaluation of that term is the crux of the matter. The accuracy of determining $Q_m A_p$ is the accuracy of determining the back losses.

The second comment, evaluation of Q_m, concerns the use of a heat meter. The point here is that a heat meter will have various readings in various places; that is, a meter would have a maximum reading if placed directly over the pipes in a panel, and would have a minimum reading if placed exactly *between* the pipes. It so happens that the average surface temperature of the panel does *not* lie exactly half way between the lowest and highest surface temperatures; therefore, the meter should

either be large enough to cover the pipe spacing (9 in.) or be placed in several locations. An example of the importance of considering the surface temperature variation can be obtained by considering the data presented in Table 1 and applying Equation 9 of the paper. Consider Test No. 3. The maximum surface temperature is 88.0 F and the minimum is 77.0 F (as recorded). According to Equation 9, $Q_m = h_r(t_s - t_a)$ and if we use 88 or 77 F for t_s then $t_s - t_a$ is 18 and 7 respectively or a variation of 257 percent. The authors do not say, but if the meter is 4 in. square, we could consider less variation, but even so the variation is of significance and should be discussed.

The last comment, the evaluation of A_p, is suggested in the paper as the difficulty of *the accurate evaluation of panel area.* From reading the paper, it was difficult to appreciate the 102 sq ft of non-panel floor area. If we consider the 102 sq ft as having some panel effect (by lateral conductance from the panel) then $Q_m A_p$ must be corrected. For example, if the 102 sq ft averaged ⅛ of the panel output (above 9 Btu per hr per sq ft) then the $Q_m A_p$ term would be increased and the back losses reduced to 10 percent.

In both items, Q_m and A_p, it is possible to take the same data and have back losses far less than the 30 percent mentioned in the paper.

AUTHORS' CLOSURE (F. W. Hutchinson): In the first question Mr. Van Nieukerken asked why the panel did not cover the entire surface of the room. It intentionally did not, because we felt that the most accurate test would be of a normal panel under a normal load. In other words, we proposed to have a panel which would function as 85 F with a water temperature of around 95 F. If we had used a larger panel it is obvious that we would have had to have a lower panel temperature.

Mr. Gordon's first question was: When were the data taken and what was the outside temperature? As indicated, we took data over an entire heating season. Most of the data were taken at approximately midnight, because that is when we usually found steady state was most assured. I cannot tell you the outside temperature range exactly, but it was approximately 0 to 20 deg.

Mr. Gordon's second point was with respect to the insulation which separated the heated section of the floor surface from the unheated section. That insulation, as the paper says, was three-quarters of an inch of cellular material. It did not extend below the concrete slab. Mr. Gordon is quite correct that some of the energy which we call a loss obviously would pass to the surrounding unheated section; we have called attention to that in the paper. You will find that we have brought out the fact that the losses as we have used the term are not necessarily all ground losses.

Question No. 3 of Mr. Gordon's also ties in with Mr. Chapman's; that is the matter of surface temperature. He points out that the five thermocouples give quite different readings of surface temperature and asks how we know that those five average the proper value? The data in the paper do not enable you to reach the conclusion that it is a true average, but I may say that before we started the tests we used an exploring thermocouple taking readings at distances of a few inches all over the entire room surface. At the same time, in each of those runs we would take off five thermocouple readings and change the position of the thermocouples so that eventually we found the five particular thermocouples which gave an average value that corresponds very closely to the integrated summed value of the exploring thermocouple readings over the entire room surface.

Question No. 4 of Mr. Gordon's was with respect to any other type of heat in the room. The answer to that is yes; the room had a ceiling panel heating system as well as a floor panel heating system, but during the period of the tests the panel under observation is the only one which provided energy.

Mr. Gordon asked whether we compared the output of the panel with our load calculations as determined by THE GUIDE method. We did not do so because such

a comparison would not have had significance for the purpose of these particular tests.

Mr. Gordon's sixth question concerns the film coefficient. Whether it should or should not be used in design is not a matter pertinent to our paper. However, the reason we developed the equivalent coefficient from this paper was primarily to emphasize the fallacy of claimed panel ratings of 80 Btu per sq ft or 90 Btu per sq ft.

In Mr. Gordon's seventh question he brought out that some of the 30 percent energy loss would have occurred even if you did not have a panel system. We certainly agree and I think that his point is an excellent one.

Mr. Gordon asks the question, as Mr. Chapman did also, of whether we should deduct the obstructed area. In the first place, we assumed there was no obstruction; we did not make any correction and we received one answer, 27 percent. Then we took very conservative assumption as to what part would flow up and we obtained an answer of about 30 percent. All we say now is that since 27 percent comes from a poor assumption in one direction, and 30 percent from a poor assumption in the other direction, the accurate answer is going to be 28 or 29 percent.

For the Q_m part, Mr. Chapman is correct. As I mentioned before, we used exploring thermocouples to find out how much variation occurred around the floor surface. In an earlier paper you will find a detailed discussion of results from the same test room showing the extent to which variation in temperature occurs horizontally between coils. I mentioned, however, that we used exploring thermocouples to determine the average temperature. In that way we selected a position in the room for standard use of the heat meter for which the surface temperature would correspond very closely to the average value of the five thermocouples.

In conclusion we note that a number of the discussors emphasize the evident need for insulation with floor panels and they suggest that the results in the present paper are not representative of average floor installations. We quite agree and want merely to remark that the purpose of our project was solely to obtain accurate experimental data from a typical *uninsulated* floor panel. Comparative data from other installations would be invaluable and we greatly hope that such data will soon be forthcoming.

1408

No. 1408

OPERATING EXPERIENCE AND DATA FROM A SIDEWALK SNOW-MELTING SYSTEM

By L. A. STEVENS* AND G. D. WINANS**, DETROIT, MICH.

SNOW-MELTING systems for driveways, sidewalks and other paved surfaces are not new. They have been used successfully for years so that there is now no question of the feasibility of such a system. Although many successful installations, larger than the one described herein, have been made, very few operating data have been available to date.

The installation of a sidewalk snow-melting system on two sides of a modern restaurant building, situated on a corner in the business district of Detroit, and supplied with steam purchased from a local district heating company, provided an opportunity to obtain valuable operating data which are reported in this paper. The system and its operation are described and data are given for both steam and electrical consumption.

In this system there are nine prefabricated pipe grids installed in the 6-in. concrete sidewalk, with about 1½ in. of concrete above the grids (Fig. 1). Concrete pads were installed at the proper grade, as shown in Fig. 2, to support the grids during the construction period. These pads are 12 in. wide by 2 in. thick and extend the full width of the sidewalk.

There are five grids on the Clifford St. side and four on the Washington Blvd. side, as shown in Fig. 2. Balancing valves and air vents were installed on each grid.

The system was filled with a solution of anti-freeze and water to prevent freezing at —20 F. Since some anti-freezes are considered toxic, all solution supply and return mains were installed in the sidewalk instead of the basement, with access boxes over the balancing valves and air vents. The heating surface provided by this piping permitted a reduction in the amount of pipe in the grids.

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Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

The supply and return piping is 2½ in. The pipe for the headers is 2 in. and the grids are 1¼-in. pipe on 12-in. centers. All joints except those at the balancing valves are welded. There are 2723 sq ft of sidewalk surface on both sides of the building. The total heating surface of the piping in the walks is 1280.

All piping in the sidewalk was covered with ¼-in. of bitumastic before the concrete was poured to permit a slight movement of the pipe in the concrete.

Expansion loops were provided in the supply and return piping, as shown in Fig. 3. These were covered on the sides of the pipe with air cell insulation before the concrete was poured to provide free movement of the pipe at these

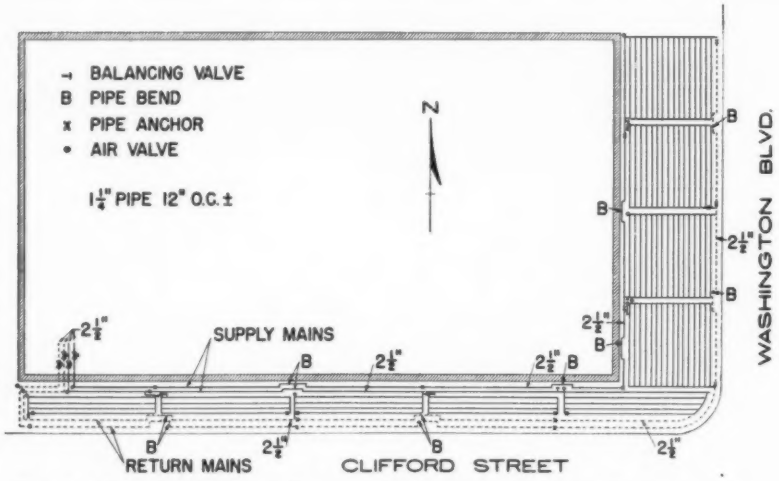

FIG. 1. ARRANGEMENTS OF GRIDS AND SUPPLY AND RETURN PIPING IN SIDEWALK SNOW-MELTING SYSTEMS

points. The air cell insulation crushes upon movement of the pipe, thereby leaving a space between the pipe and concrete. Reinforcing wire also was installed over the piping before the concrete was poured.

Fig. 4 is a diagram of the equipment that is installed in the basement for each of the two systems. Before erection, the equipment was assembled on the floor. This assembly is shown in Fig. 5. The equipment was the same for each of the two systems and includes the following:

1. Heat exchanger having a capacity of 8450 gal per hour with a 20-deg temperature rise using steam at atmospheric pressure.
2. Expansion tank, 13 in. in diameter by 50 in. long.
3. Water pressure relief valve to discharge to floor drain.
4. Water circulating pump equipped with a ½-hp motor.
5. Solution temperature control valve on the steam supply.

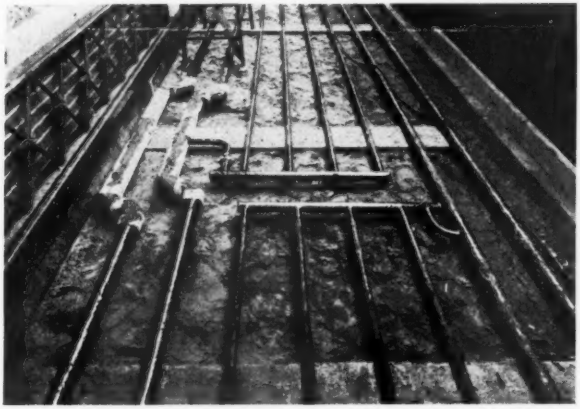

FIG. 2. GRIDS IN PLACE. SHOWING BALANCING COCKS AIR VALVES AND THE BITUMASTIC COATING

Anti-freeze is put into the system at openings in access boxes in the sidewalk. A removable hose connection, as shown in Fig. 4, is used to add water, because permanent connections between city water supply piping and snow-melting systems are not permitted.

Details of the operation of the snow-melting system during two snowstorms follow:

Snow began falling on January 6, 1950, at 5:45 a.m. and at 6:30 a.m., after some snow had accumulated on the sidewalks, the snow-melting systems were turned on.

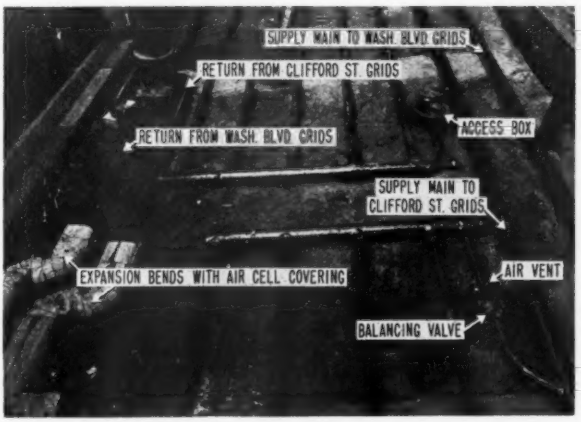
SUPPLY MAIN TO WASH. BLVD. GRIDS
RETURN FROM CLIFFORD ST. GRIDS
ACCESS BOX
RETURN FROM WASH. BLVD. GRIDS
SUPPLY MAIN TO CLIFFORD ST. GRIDS
EXPANSION BENDS WITH AIR CELL COVERING
AIR VENT
BALANCING VALVE

FIG. 3. BALANCING VALVE AND ACCESS BOXES

At 9:00 a.m., although the sidewalks were wet, all snow was melted. Snow continued until about midnight for a total snowfall of 4 in. The systems were turned off at 12:30 a.m. Outdoor temperature on January 6 ranged from 25 to 28 F.

A total of 9550 lb of steam was consumed during the 18-hour period. The maximum hourly use was 1013 lb, or 0.37 lb per sq ft of sidewalk surface. The total electric consumption for the period was 6.5 kwhr. These results were satisfactory to the user, but would have been more so if the system had been started when the snowfall began. Supply and return solution temperatures for the Clifford St. system were 135 and 125 F, respectively; and for the Washington Blvd. system 135 and 120 F, respectively.

On February 13, 1950, there was a 5-in. snowfall starting at 4:45 a.m. and continuing until 12:30 p.m. From 12:30 to 1:00 p.m. freezing rain and sleet fell, and then rain started falling at 1:30 p.m. By midnight of the 13th, 1.88 in. of sleet and rain

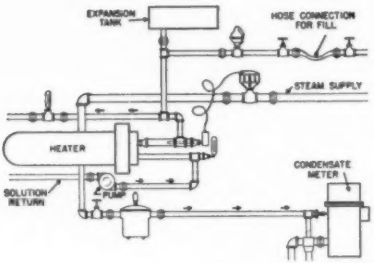

FIG. 4. DIAGRAM OF SNOW-MELTING EQUIPMENT AT HEATER

had fallen. From midnight of the 13th to midnight of the 14th, there were 1.92 in. of rainfall.

At 6:15 a.m. on February 13, two hours after the snowfall began, the snow-melting systems were turned on and left on until 8:30 a.m. of the 14th — a total time of 26¼ hours.

During the first part of the period, when snow or sleet was falling, the outdoor temperature varied from 26 to 30 F, and did not get above freezing until after midnight of the 13th. At 9:00 a.m. of the 14th the outdoor temperature was 34 F. Had the system not been in operation, it is probable that the rain would have frozen on the sidewalk. The total steam consumption for this period was 13,400 lb. The maximum hourly steam consumption was 845 lb, or 0.31 lb per sq ft of sidewalk surface. The total electric consumption of the two water circulating pumps was 11 kwhr for the 26¼-hour period. Here again, the performance was satisfactory to the user, but would have been more so if steam had been turned on earlier.

The curves in Fig. 6 indicate hourly steam consumption data for the entire building as well as for the two snow-melting systems during this period.

On another day when there was no precipitation, the systems were operated for 9 hours to determine the steam consumption under these conditions. The systems were started at 6:00 a.m. with an outdoor temperature of 17 F and operated until 3:00 p.m. when the temperature had risen to 36 F. The temperature of the solution being circulated was 100 F.

The total steam consumption for the 9-hour period was 2350 lb with the maximum hourly use being 340 lb, or 0.125 lb per sq ft of sidewalk surface. The electric consumption for this period was 7 kwhr.

The amount of pipe surface and the size of other equipment in a snow-melting system could be reduced if the system were operated continuously. The system in this report was an actual operation for 155 hours for the entire season, which we may consider from November 15 to March 31, or 3288 hours. It would appear that, if a system were operated continuously, about 3100 hours of this operation would not be necessary. From the test figures above, this

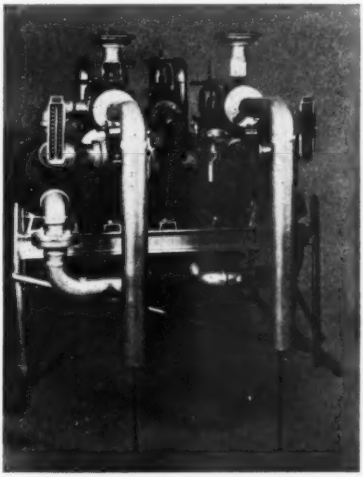

FIG. 5. ASSEMBLY OF EQUIPMENT BEFORE ERECTION

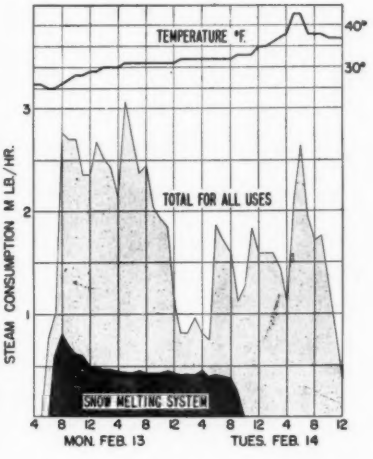

FIG. 6. HOURLY STEAM CONSUMPTION DATA

would mean 3100 × 0.1250 lb = 387 lb steam per sq ft of sidewalk would be wasted.

The total snowfall in Detroit during the winter of 1949-50 was 40.1 in. The total steam consumption of the snow-melting system was 97,947 lb and the electric consumption 64.5 kwhr, as shown in Table 1 for the winter of 1949-50.

At another restaurant in Detroit, owned by the same company, it was necessary to have a man at the entrance to mop up the slush tracked in by patrons. This was not necessary at the new building provided with the snow-melting system.

Some additional surface should have been installed under the revolving door since, during the snowstorms, a small amount of snow collected just outside of the revolving door.

No provision was made to drain the melted snow from the sidewalk. The water evaporated about as fast as the snow melted.

TABLE 1—SUMMARY OF DATA FOR SNOW-MELTING SYSTEM
(Winter 1949-50)

	Hours of Operation	Steam Consumption Pounds	Electric Consumption Kwhr	Snowfall Inches	Remarks
	25½	13,175	10.0	2.6	Steam on beyond snowfall
	3½	2,702	3.5	1.1	Steam on during snowfall only
	6½	4,764	2.0	2.0	Steam on during snowfall only
	4¾	3,864	3.0	2.0	Steam on during snowfall only
	1¼	1,539	0.5		No snow—threat of snow
	2½	2,383	1.5	0.7	Steam on during snowfall only
	18¾	9,550	6.5	4.0	Steam on beyond snowfall
	¼	244		trace	
	1¾	1,538	1.0	trace	
	3¼	2,515	1.0	trace	
	5½	3,715	3.0	2.7	Steam on during snowfall only
	11	6,624	4.0	2.6	Steam on beyond snowfall
	26¼	13,401	11.0	5.0	Steam on beyond snowfall
	½	582		trace	
	8¼	5,948	1.0	2.0	Steam on during snowfall only
	16¾	10,209	8.0	3.6	Steam on beyond snowfall
	4	2,928	2.5	2.0	Steam on during snowfall only
	2¼	1,837	1.0	0.6	Sleet, rain, showers, and snow
	4	3,094	1.0	4.6	Steam on during snowfall only
	4¾	3,377	3.0	2.1	Steam on during snowfall only
	4	3,956	1.0	2.5	Steam on during snowfall only
Total	155.25	97,947	64.5	40.1	

CONCLUSIONS

Some conclusions regarding installation and operation of snow-melting systems based on this installation are:

1. There is no need to provide drainage for the melted snow.

2. Solution heaters should be capable of condensing 0.4 or 0.5 lb of steam per hour per square foot of sidewalk surface.

3. The steam and electric consumptions are reasonable.

4. Although not determined in this test it is obvious that where salt is used to melt ice and snow on sidewalks bordered by grass or shrubs, a snow-melting system would eliminate the need of the salt which would be injurious to plant life.

5. It is desirable to operate the snow-melting systems only when there is snow on the walk or when snow is falling.

6. There is a need for a simple control which would start operation of the system when a snowfall begins and turn it off when the snowfall is ended and the sidewalks are free of snow.

7. A snow-melting system insures cleanliness in entrances and lobbies during snowfalls.

ACKNOWLEDGMENT

The authors extend their appreciation to all who have participated in the design and installation of this snow-melting system and in the preparation of

the paper, including, A. J. Caughey, heating engineer, The Stouffer Corp., the contractor, Galvin Plumbing and Heating Co., and its foreman, John C. Ritter, and to C. Howard Crane and Associates, Inc., architects.

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DISCUSSION

L. J. LATART, Rome, N. Y., (WRITTEN): Commendation is due the authors for the excellent information published in this paper. The steam and electric consumption rates given will prove helpful to designers in estimating operating costs of snow-melting systems. It is interesting to note the large savings that can be effected from intermittent operation of the system, since the required operating time is shown to be such a small portion of the total hours in the winter. It would be of further interest to know the average snowfall for this area.

During the last three winters, we have been experimenting with three snow-melting systems. Two of the systems are installed in *black top* walks and driveways, and the third system is installed in a *concrete* loading ramp. We have found in the *black top* installation that tube spacing of 6 in. to 9 in. center to center, depending on length of coil circuit, is desirable to produce good melting patterns. There seem to be two reasons for this, first, because of the thermal conductivity of the black top and second, because the depth of bury of the heating coil is generally from 1 to 1½-in. below the surface.

We should like to ask the authors the following questions: (1) What was the pattern of melting when starting with a cold slab after several days of cold weather, and after snow had accumulated on the slab? (2) Under those conditions would slushing occur between the rows of pipe during the melting period? (3) What was the distance between the expansion loops in the piping system? (4) What was the distance between expansion joints in the concrete walk? (5) Have any noticeable cracks developed on the surface of the concrete?

G. LORNE WIGGS, Montreal, Can.: I was very much interested in this paper. In Montreal we have the problem of melting a snowfall of 100 in. or more. I note that the heaters for this sytsem were designed to give 520 Btu per sq ft of sidewalk per hr. To melt the snow requires about 260 Btu per (sq ft) (hr); in the figures obtained the authors indicated 280 Btu per sq ft.

I wonder why there are two systems of heaters and pumps, as one set would give a much lower installation cost.

The authors mention the fact that no provision has been made for draining away the water. Where valves are imbedded in the sidewalk, as in this case, might not the enclosures or the spaces around the valves become filled with water and freeze when the system was shut off?

In this case considerable thought has been given to the expansion of the pipes. We have been burying them in the concrete without provision for expansion. In fact, we have had coils up to 80 ft long without cracking of the concrete.

The pipe described in this paper seems large for the purpose. We think the ½-in. or ¾-in. pipe will perform equally well but in most cases we use ¾-in. pipe, in order to keep the friction drop down to satisfactory limits. Certainly ¾-in. pipe is appreciably cheaper to install than 1¼-in. pipe and requires much less anti-freeze solution.

On the matter of spacing the pipes and the bury; the authors used a bury of 1½-in. and a spacing of 12 in. You would find in a heavy snowfall of 6 in. or more that with such light bury and spacing you would get a series of furrows. You might even melt the snow so it is tunneled over the pipes and yet the top surface would remain intact. This furrowing or tunneling can be eliminated by reducing the spacing between the pipes and also by greater depth of bury, which will give a more uniform temperature of the surface.

P. B. GORDON, New York, N. Y.: I would like to ask whether the 400 or 500 Btu per hr per sq ft of sidewalk area that is indicated for the capacity of the solution heater was recommended for the pickup load, or whether it was intended to take care of the melting rate only. Many jobs are now operating in this climate with heat inputs in the order of 130 to 150 Btu per hr per sq ft. I am curious to know the reason for the 400 or 500 figure mentioned in the paper.

F. H. BUZZARD, Philadelphia, Pa.: I note in the hookup that the City Fathers permit a makeup connection using a hose without even a check valve; yet they required valves in the sidewalk. I am wondering why they did not require an open expansion tank and prevent any possibility of the solution getting back into the water system.

AUTHORS' CLOSURE: The authors appreciate the comments. In answering Mr. LaTart's question, the average snowfall in Detroit is about 40 in. In the winter 1949-50 the total snowfall was 40.1 in. The patterns are exactly the patterns of your grid. You will start out melting directly above your piping and continue outward as time goes on if you start from a cold slab. Of course, if you start from a warm slab and input is sufficient, you will have no pattern; you will just have a wet sidewalk.

Slushing did occur when you started up the system after snow had fallen. It would be well if you could anticipate or if you would start your snow-melting system as snow begins to fall. You would have no slushing and we believe the system would be more satisfactory to the customer.

Some rather small cracks have appeared. These are not any greater in our opinion than might be expected in a walk without snow-melting. The distance between pipe expansion loops is about 40 ft. As I recall it, the expansion joints in the sidewalk are about eight feet apart.

Mr. Wiggs' comments are very interesting and with a snowfall of 100 in. you really get some work. We discovered in one experimental system that we tested,

1000 Btu were consumed per sq ft of sidewalk per hr. Considerable of that might have been edge losses because it was a small panel. Actually, in this system we were using 0.37 lb of steam per sq ft of sidewalk per hr, or roughly 370 Btu.

As for the duplication of the system, this building was built without too much consideration given to cost; dependability being the main purpose. Surely, if it were built today only one system would be used.

The access boxes made of bronze are set in the sidewalk and have screw tops. It is not likely very much water will get inside of them. Tunneling and furrowing is very possible, but, as Mr. Wiggs points out, that can be corrected by deeper burial and putting your pipes closer together. The hose connection is permissible only during the filling of the system, and must be disconnected as soon as the system is filled.

Mr. Avery brings out the point that the melting of snow on airplane runways is probably very important. There have been some attempts at it.

Incidentally, it might be of interest to mention that we tried at one time to lance the snow from the sidewalk with a steam jet, but that was not too feasible, either from lack of knowledge or persistence.

1409

No. 1409

DESIGN FACTORS IN PANEL AND AIR COOLING SYSTEMS

By C. S. LEOPOLD*, PHILADELPHIA, PA.

THE PERFORMANCE of a cooled panel is better than would be anticipated by the calculations used for panel heating, as the major internal sensible loads for which cooling is required appear in significant proportion as short wave length radiant energy, which is usually not recognized in panel heating. Since the air is substantially transparent † to radiation, it is not directly warmed by radiation. Radiant energy may be converted into thermal energy when it strikes a solid. The amount of radiation reflected and the amount converted into thermal energy depends on the absorptivity of the surface, the difference in the fourth powers of the absolute temperatures of the source and sink, and their geometry. For sunlight, and for large portions of the energy emitted by a luminaire, there is such a large difference between the temperatures of the source and the structure that the conversion of radiant energy to thermal energy is practically independent of the surface temperature of the structure. Where radiant energy impinges upon a cooled panel, a portion of the radiation which is converted to thermal energy is removed even though that panel be at or somewhat above the room air temperature. The portion of radiant energy which can be so removed has been measured and named Independent Radiant Transfer[1] (IRT).

The radiant energy absorbed by the surface of walls, floor, or uncooled portions of a ceiling is, in part, conducted into the structure and, in part, transferred to the air by convection or to other room surfaces by re-radiation.

In conventional systems using only air for cooling, it would be expected that surfaces not provided with means of cooling would tend to rise above the room air temperature. The fact that the surface of the structure is directly warmed by absorption of radiant energy would also indicate that with a conventional cooling system there will be an increase in thermal storage even though cool air is supplied in quantity to maintain a constant room air temperature.[1,2]

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* Consulting Engineer. Member of A.S.H.V.E.

† *Heat Transmission*, by W. H. McAdams (2nd Edition, pp. 64-70).

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

[1] Exponent numerals refer to Bibliography.

TEST DATA

Data have been presented [1] on the performance of a continuous, cooled metal ceiling, with various types of luminaire, in a test structure controlled so as to simulate a condition of thermal equilibrium between room air and structure.

Data have been presented [3] for the solar load in a room with a continuous, cooled metal ceiling, a 10-in. cinder concrete floor, and with various arrangements of glass and shading devices. Tests were conducted to simulate the effects of thermal storage, with the under side of the floor slab maintained at a constant temperature corresponding to design room air temperature.

For these tests, the room and fenestration were represented by a scale model of the building then being studied and solar radiation was simulated by a combination of a mercury vapor lamp and filament reflector spots.

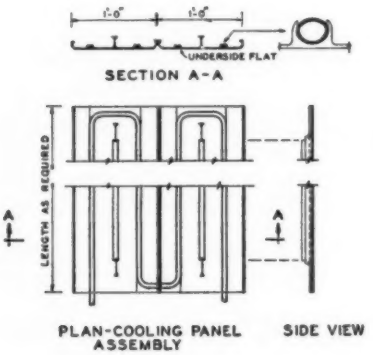

FIG. 1. COOLING PANEL DETAIL

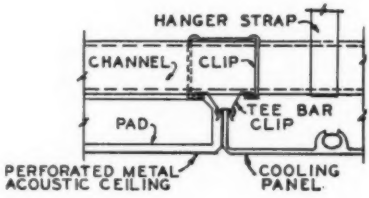

FIG. 2. METHOD OF SUSPENSION

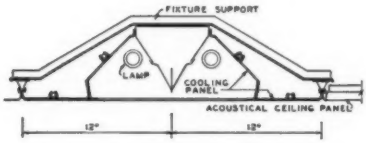

FIG. 3. TROFFER LUMINAIRE PANEL

Appendix A contains subsequent data on panel performance for a suspended ceiling consisting, in part, of cooling panels and, in part, of perforated metal in conjunction with sound-absorbing pads.

For the test with luminaires, 29 percent of the suspended ceiling was cooled. Tests were conducted with and without the storage effects of a 10-in. cinder concrete slab.

For the tests with solar heat and shading devices, 40 percent of the suspended ceiling was cooled. The experiments were otherwise conducted as previously described.[3]

PANELS

General: Theoretical considerations corroborated by the tests previously outlined indicated that it was not necessary to cool the entire ceiling in order to utilize coolant temperatures safely above the dew point.

Applied sound-absorbing materials are generally of low thermal conductivity.

Cooling pipes imbedded in the actual ceiling structure would be effective but are open to some objection on the basis of complicating the application of acoustic treatment and in that they would require the coolant to be at lower temperature than would be required with a metal ceiling because of the temperature drop, coolant to surface.[4] Despite this limitation, cooling by tubes imbedded in plaster or concrete is feasible.

Cooling panels can be perforated to allow for sound-absorbing media above without materially affecting their thermal performance but a greater economy in the use of materials may result when only a portion of the ceiling is in the form of panels and the remainder in less expensive materials, such as conventional metal pan acoustic ceilings.

One System: For the acoustic treatment of the average office, half of the ceiling as perforated panels will usually be sufficient. The general acceptance of acoustic treatment in modern office buildings led to the thought of introducing

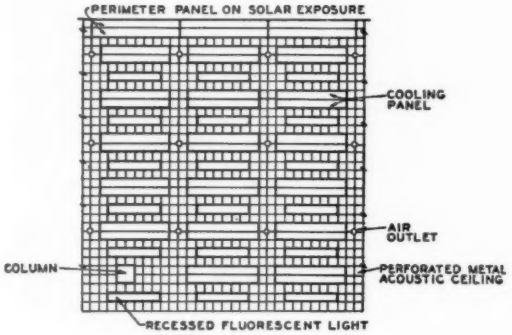

FIG. 4. REFLECTED CEILING PLAN

the cooling panels and acoustic panels in a form which would fit the ordinary suspension for a perforated acoustic ceiling. Cooling panels were designed as shown in Fig. 1. The basic material is an aluminum extrusion, approximately 0.062 in. thick, with channels on the top in which copper tubes are placed in close fit and then deformed to provide thermal contact. Tubes can be deformed either by flattening or by expanding. Copper tubing was selected because of its ease of assembly and its known resistance to corrosion. The panels fit into a typical ceiling suspension as shown in Fig. 2. They can be arranged in pattern as desired and, if it were necessary, the cooling panels could be perforated for additional acoustic treatment.

Cooled Luminaire: Exploiting the idea of removing as much energy as possible at the source, led to the design of the panels in part as the reflector for a lighting fixture, as shown in Fig. 3. The performance per unit area of such a panel is higher than a normal ceiling panel since much of the radiation, as well as convected and conducted heat, is directly intercepted before it warms other room surfaces or the room air.

With any type of cooling panels forming a part of a suspended ceiling, the upper surface of the panel receives heat energy from the air above and by radiation from the under side of the floor slab or roof above. The air above a suspended ceiling is warmed by heat conduction through the ceiling structure which is not directly cooled and may be warmed by auxiliaries of the lighting system. There is some conduction from the uncooled portion of the ceiling to the cooled panel, in amount depending upon materials of construction of the uncooled ceiling and the conductive properties of the bond.

Use of the luminaire panel is desirable where recessed fluorescent lighting is to be used. Where applicable, it has the mechanical and aesthetic advantage of reducing the number of different types of elements which form the ceiling. With flat plates, unless the plate is perforated to match the ordinary cheaper materials of construction used for the acoustic pans, there are four elements;

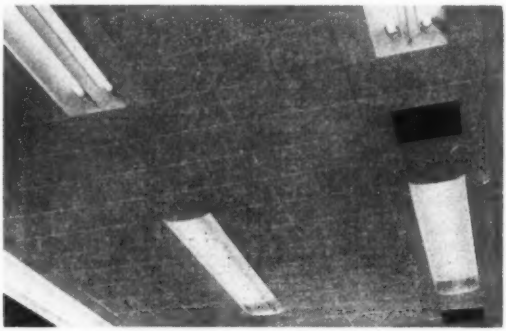

FIG. 5. CEILING CONSTRUCTION, MANUFACTURERS LIFE INSURANCE BLDG.*, TORONTO

namely, the air grille, the acoustic pans, the cooling panel and the luminaire. With the cooled luminaire, the identity of the cooling panel is lost, thus removing one element. This is important in design as it is easier to obtain symmetry with the three elements than four.

In general, the design of a suspended ceiling panel cooling system will require from 20 to 40 percent of panel. Fig. 4 shows a reflected ceiling plan using recessed fluorescent lighting and flat panels. Fig. 5 is a sample ceiling using the luminaire cooling panels.

Below Window: The flat panel can also be used as a heating and cooling means below the window, and one method of application is shown in Fig. 6. The panel is used for cooling in summer and provides a cool surface to counterbalance the radiation from a hot window or venetian blind.

HEATING

In winter, the panels below the window can usually be the sole means of heating. Although the panels on the ceiling adjacent to windows can be used

* Architects: Marani & Morris; Engineers: Charles S. Leopold, and Wiggs, Walford, Frost & Lindsay.

for heating, it is believed that optimum conditions are better obtained by providing a warm surface on the outside wall directly below the cold glass, where it serves both to offset the radiation and to minimize the flow of cold air down the glass. These panels have the additional advantage that during the night they are the only part of the system requiring operation to keep the temperature of the structure balanced, thus reducing the required period of operation of the entire system.

The foregoing description covers suggested uses of extruded aluminum panels. There are, obviously, many other ways of cooling a large surface by

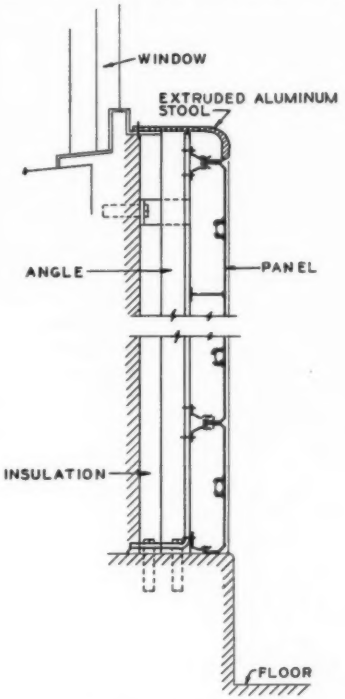

FIG. 6. WALL PANEL DETAIL

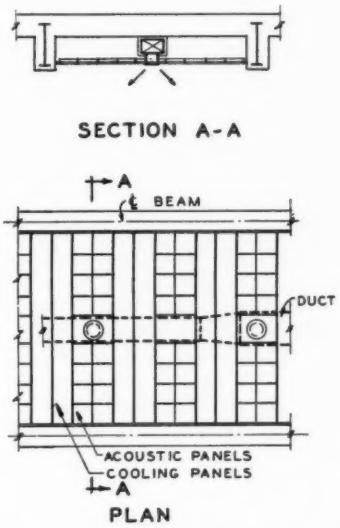

FIG. 7. REFLECTED CEILING PLAN

a tube carrying a coolant as, for example, by thermally bonding metal tubing to rolled panels of any material or, as previously noted, by imbedding the coolant tubes in a plaster ceiling or in the concrete of the construction. The selection of the type of ceiling and the area to be cooled would be determined by economics and the need for acoustic treatment.

VENTILATION

For this discussion, forced ventilation has been assumed. Where it is desired to obtain relief, rather than controlled summer comfort, panel cooling can still be effective but the coolant temperature must be controlled to be above the dew

point at all times. The conventional cooling means, with control, can be used or the coolant can be re-cooled by evaporation, thus insuring that the temperature at all times will be above the dew point. For controlled comfort cooling, air should be introduced in the amount and at a dew point to produce the necessary drying effect and ventilation and, since the dehumidification of the air is usually though not necessarily accomplished by refrigeration, the air may be economically used to produce a portion of the cooling effect. As previously pointed out [1], any thermal energy removed by the air has more than a proportional effect in reducing the required temperature difference between the room air and the panel, or in reducing the amount of panel required to remove the remaining sensible load.

SPACE REQUIREMENTS

The reduction in required volume of air supply, whether it be delivered at high velocity or at more conventional velocity, results in a simplification of the air distributing problem, with lesser requirements for ceiling heights than conventional systems.

The design contemplated for a building which was to have 30-in. beams to span approximately 40 ft, is shown in Fig. 7. It was found that even with a building of large floor area, the largest hole required in the 30-in. beams, for horizontal distribution at low velocity, was 10 in. $\times$ 14 in. and the installation could, therefore, be made without effecting ceiling height. The acoustic and panel ceiling could be kept above the lower edge of the fire-proofed beam.

Panel cooling is well suited for buildings having large interior zones and may have special application in buildings which employ other methods but have separate areas of illumination requiring high energy levels.

The rentable area lost by air conditioning equipment is at a minimum because substantially the only floor space required outside of the apparatus room is that for the vertical trunk ducts when the major distribution is vertical. These ducts are inherently small because of the lower air quantity and may be designed for either commercial or for high velocity as the economics indicate. In general, the penalty incurred with high velocity in a trunk duct is very much less than in using high velocities in branch ducts.

The conservation of rentable area and the clear floor plan is the more apparent advantage of panel cooling. Its relative thermal stability is also significant and will be discussed.

ZONE *vs.* INDIVIDUAL CONTROL

As an end result, the purpose of comfort air conditioning is to adjust the thermal environment so that the occupants can work without being conscious of discomforts or annoyances due to air temperature, humidity, odors, air motion, and radiation.

For a group, the optimum temperature and humidity is well established. For an individual, there is no such assurance. The group optimum is a statistical average and any individual may depart slightly from the statistical optimum. It follows that where a number of people must work together in an enclosure, a constantly maintained optimum is desirable. Means of individual control for maintaining a different temperature, as between two rooms, both of multiple occupancy, or two parts of a large room, is not required. If two or more people

are constantly working in one room, they tend to require the statistical optimum and individual control is usually not required to maintain conditions other than the optimum. Depending on design, it may be required to maintain the selected optimum.

In a single occupancy private office, there may be a demand for separate temperature control, either because the occupant wishes a small adjustment representing his departure from the statistical average, or because he wants to feel cold or warm.

The purpose of this discussion is to show that there are two reasons for a multiplicity of controls or of control by small area. The first, if because of design it is necessary in order to maintain the selected optimum for the group; and the second, where it is desired to maintain a special condition for an individual which would not be acceptable to the group. Individual control is generally costly and can frequently be avoided by proper design and control of a zoned system. In this latter case, the determining factor is how well the system can meet different loads in adjacent areas of the same zone. If one room has more people and lights than an adjacent room with substantially the same air supply, how far will they differ in temperature? For years there have been installed zone systems in which air is introduced at from 12 to 25 deg below maintained room temperature. For two rooms side by side; one occupied—lights on, one unoccupied—lights off, there would be anticipated a marked difference in temperature. In fact, if no factors such as heat storage were taken into account, the one room could be assumed to differ from the other by the amount of the temperature difference of the air introduced and the general average of air temperature in that zone. Actually, it is found that these differences do not materialize to this degree. This helpful phenomenon has been accepted and assumed as a result of thermal storage but storage, until recently, was considered as being caused by the warm air heating the structure by convection. It is now known that this effect is greatly accelerated by radiation and that the structure is frequently warmer than the room air. An overloaded room, for example, does not get so warm as one would expect because radiation has accelerated the rate of storage.[1,3]

An approximation of these results by analogue is shown in Appendix B. In Fig. B-5 a comparison of Columns 1 and 3 and Columns 2 and 4 indicates that, in addition to the effect of radiation and convection on storage, the actual conduction through floor and ceiling is significant and that, as an end result, the variation between the underloaded and the overloaded room, with air introduced 26.3 deg below room temperature, is 9.7 deg (Column 4); whereas by conventional calculation without regard to storage, a 23.2 deg variation would be indicated. With air introduced 13.3 deg below room temperature, the deviation is 5.6 deg (Column 3) against a calculated 11.7 deg. The consideration of walls would further limit the deviation when factors of radiation, storage, and conduction are considered. These results explain the satisfactory behavior of well designed conventional zone systems and also show why poorly designed systems sometimes perform better than would be expected by conventional calculations.

Since a very narrow deviation of maintained temperature is acceptable, need for large air quantities and low temperature differences between the introduced and room air is still of importance for conventional systems. Unfortunately, large air quantities frequently result in the requirement for additional building

areas or cube and, although this is seldom a determining factor in smaller buildings, it can be and often is of major significance in large buildings. This limitation is frequently not so serious in an existing building with higher ceilings than are customary in new construction.

The response of a panel cooling system to the same conditions of loading is shown in Fig. B-7. For the stated condition, the deviation between the overloaded and the underloaded room is 4.3 F as compared to 7.0 F with the conventional 13.3 T.D. system, as graphed in Column 1 of Fig. B-5. These results were obtained by simulating a completely rather than a partially paneled ceiling, and again do not take into account the effect of walls. In interpreting these results, consideration should be given to the following: They apply to an interior zone and to a load due principally to luminaires. In an office building, this is the preponderant load. For the solar load it was shown[3] that with a given masonry opening and for all types of glass, single and double, and for commonly used shading devices, room air temperature could be maintained by depressing the ceiling temperature by an amount varying from 12.1 to 9 deg, a difference of 3.1 deg.

The foregoing indicates a second advantage of panel cooling; namely, the possibility of maintaining close to the predetermined optimum with lesser need for individual controls than would usually be found in the ordinary well designed conventional zone system.

Individual controls can, if desired, be applied in the form of air volume, air temperature, or by control of water quantity or temperature to panels in a given area. Unless individual control is desired to take care of a single occupancy office, the air may be delivered at a predetermined constant temperature and the variations in load may be met by varying the water temperature to the panels by zones. There is considerable advantage in using constant temperature air supply to all zones of the building in that it avoids a multiple duct system with its complication of duct crossings. The trunk ducts become smaller in total area as higher velocities can be used in a larger duct at the same power expenditure. Constant quantity and temperature air delivery has proved satisfactory in an experimental installation over a three-year period of operation.

CONTROL IN A TEST INSTALLATION

Three rooms with a south exposure and one adjacent room in the interior, on the 32nd floor of a New York office building, were equipped with a test installation of panel cooling. The air supply to the south rooms was 0.5 cfm per square foot and to the interior room was 0.4 cfm. These air supplies were maintained at 60 F summer and winter, with a dew point at a maximum of 50 F. The conditioned space was completely surrounded by unconditioned space and a larger percent of the ceiling was in the form of panels than would be required in a completely conditioned building. Control of the panels was as follows:

The panels beneath the window sill were controlled by a thermocouple on the inside of the windowpane shielded from the direct rays of the sun by a metallic disc on the outside of the glass. This thermocouple regulated the supply water temperature for the panels below the windows. Since the major loss or gain of heat by transmission is through the glass, this device, to a large degree, integrated the effect of wind velocity, outside temperature, and, to some extent, the effect of the venetian blind. Thus, if the glass is at a temperature of 75 F, there is no need for either heating or

cooling of the sill panel and this condition could prevail even on a cold winter day, if there is enough energy from the sun. The area of such panels is limited so that generally they cannot provide for the full solar effect.

The temperature of the water to the ceiling panels directly adjacent to the windows was controlled by the intensity of the sunlight as measured by a photo-electric device located outdoors. As the solar intensity increased, the water temperature was lowered. Photo-electric control, plus shielded outside dry bulb control, has proved satisfactory over a period of years for the primary control of air temperature to low pressure window units in a very large structure.

There was question as to whether the photo-electric device or an instrument similar to a pyroheliometer should be used. Tests with a heat meter and a light meter indicated that there was a straight-line correlation between total and visible radiation, with the exception of a few isolated deviations which were so few in number that the complications of maintenance of a pyroheliometer were considered unnecessary. The advantage of the photo-electric meter is that it is rugged, sensitive, commercially developed as a control means and, further, that dirt has little effect on its operation as it must be initially shielded with thick colored glass.

It will be noted that these controls respond to the outside conditions before they have had a chance to produce a measurable effect on the air temperature within the building.

The interior zone in the test installation was intended to be controlled by a time clock providing predetermined temperatures in accordance with a schedule. It was found, however, that one temperature of water was sufficient year-round. This installation differs from a fully air conditioned building in that the four rooms under test were surrounded by unconditioned space. Some elaboration of interior zone control is anticipated in a fully air conditioned building. The suggestion of a time clock to set a predetermined schedule of water temperature should be satisfactory, or it would be possible to control the temperature of the water to the interior zone panels from the changing electric power demand of that zone. In this way, the cooling effect would be applied as soon as there was load and would therefore anticipate and integrate the cooling requirements.

This control system recognizes that with a zone system there is little that can be done in sensing minor variations in load as between areas in the same zone and that the primary response of the system would, therefore, be better obtained by directly sensing the major load and making an allowance for the minor energy gains. The controls are applicable to zoned panel or air systems.

With conventional controls, it is customary to locate a thermostat either in an area considered to be average or in a return air duct. There are difficulties encountered with either of these procedures, though the method is useful. Instruments responsive to maintained air temperature all have the disadvantage of failing to call for a change until the room air temperature has been affected.

COMFORT

The comfort results of a panel cooling system are comparable to the results which would be obtained with a good conventional zone system with relatively small difference between cooling air and maintained room air temperature and with air distribution as by a perforated ceiling.

In the test installation, there was no significant increase in the optimum room temperature in summer. The rooms were held at approximately 75 to 76 F. In winter, the people who lived with this test installation reported an apparent increased tolerance for lower temperatures than would normally be expected and this was partially substantiated by the fact that there was no comment when

the temperature was deliberately varied from 75 to 70 F without the knowledge of the occupants. If the observation is correct, it may in part be explained by the following:

Temperatures are constant from the floor surface to within about 6 in. of the ceiling where a drop in temperature is noted. There is no noticeable air motion or draft, even at the floor. The sill panel provides symmetry of radiation. This reported result is the more unusual because the interior surfaces tend to be at a lower temperature than they would be with a conventional air system.

These opinions and observations are presented as a subject for further study. They cannot be considered as a definite conclusion drawn from a properly controlled test.

LIMITATIONS

There does not appear to be any problem in operating a panel cooling system with the lowest panel coolant temperature well above the desirable dew point for offices and similar structures but careful consideration must be given to applications in which it is possible for the dew point to rise above approximately 63 deg for a long enough period to produce condensation. Other than this problem of condensation, the limitations of a panel cooling system are those inherent in any system where there is a limited supply of ventilation air and the sensible cooling is accomplished by multiple cooling units each serving a small area. In a liberally designed conventional air system, from two to four times as much air is circulated as with most split systems. Although the absolute quantity of outdoor air is no greater in a conventional system, the total supply to any one office, including recirculated air from other areas, can be considered as substantially smoke-free when compared with the air in a room in which people are smoking, so that the control of concentrated sources of odor is not so good as it would be for a comparable conventional system.

The other limitation of split systems in general lies in the fact that refrigeration, rather than outdoor air, is required for more hours than would be needed for a liberally designed all air system. This effect may, in part, be offset by the relatively small power requirements of the panel system. In cool weather, with panel design, the incoming outdoor air can be caused to cool water which is to be circulated in the panels and, since the total air circulated may be greater than the minimum outdoor air requirements for peak summer weather, some extension to the hours of cooling without refrigeration may be obtained.

In applying conditioning to large buildings, there are frequently special areas which can be best served with a conventional system and, if there is a series water circuit, the outdoor air passing through these conditioners can also be caused to cool the panel water. Evaporative cooling may be used to cool the water which is circulated to the panels. In a panel system, the problem is generally easier than with coil systems since the panels will seldom require water below 65 F. With the panels, there is the further possibility of operating two levels of refrigeration, one for panel cooling and the other for dehumidification.

CONCLUSION

The author has attempted to present the theory of panel and conventional air cooling systems and to indicate possible courses of panel cooling design.

In comparing air conditioning methods, it is essential that the methods under comparison shall not produce an end result which will unduly compromise with the production of optimum conditions.

Assuming that the air conditioning methods to be compared are capable of attaining the same end result, the selection of a particular form of air conditioning is a matter of economics. The air conditioning design should be related to all elements of building construction and use, and the economics be determined not solely on the owning and operating cost of the air conditioning but on the owning and operating cost of the entire building.

ACKNOWLEDGMENT

The author wishes to express his appreciation to The Bohn Aluminum & Brass Corp. for its cooperation in the commercial development of the panel; to *Time, Life, Fortune* and *Architectural Forum*, who financed the laboratory work; and to Frank Wilder, who had charge of the experimental program for the author.

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APPENDIX A

COOLING PANEL PERFORMANCE WITH LOAD DUE TO LUMINAIRES

Part Ceiling—Steady State

The test cubicle previously described [1] was modified by the addition of a suspended ceiling 9 in. below the continuous cooling panels, as shown in Fig. A-1. The supply air grille was lowered to maintain 12 in. from the top of the grille to the under side of the suspended ceiling. Standard perforated steel pans with acoustic pads covered 71 percent of the ceiling, and 29 percent was in the form of aluminum panels of the same construction as those used in previous tests. These panels were 6 ft 8 in. long, 2 ft wide, and were separated by 2 ft of the perforated steel ceiling pans. A 1 in. layer of insulation board was placed on the under side of the original continuous ceiling cooling panels. During this series of tests the original ceiling panel was maintained at the same temperature as the surface of the insulated floor and, since the 1 in. of insulation board has approximately the same thermal resistance as 10 in. of cinder concrete, this procedure approximated the steady state condition in the interior zone of a multistory building with 10 in. cinder concrete slab construction.

TABLE A-1—SUMMARY PART CEILING—STEADY STATE

(Air Supply Where Used, 37.4 cfm—Floor Area, 92 sq ft)

LAMP TYPE	AIR SUPPLY TEMP F	PAINT USED ON PANELS[a]	PANEL TRANSFER[b]		PANEL TEMP[c] F	AIR TEMP[d] 5 FT LEVEL F	FLOOR TEMP[e] F	AIR TEMP 2½ IN. BELOW PANEL CENTER[c] F	AIR TEMP 4 IN. ABOVE PANEL CENTER F	STRUCTURE TEMP DIRECTLY ABOVE PANEL CENTER[c] F	APPARENT "b" FOR PANEL BTU/(SQ FT) (HR) (F DEG)
			BTU/(SQ FT) (HR)	% of LAMP LOAD							
500 w, Bowl Silver Indirect	No Air	#1	51.2	80.0	61.6	78.0	79.5	77.6	72.9	71.7	3.12
500 w, Bowl Silver Indirect	60	#1	28.3	44.0	69.4	78.0	79.2	77.5	74.0	73.8	3.29
500 w, Bowl Silver Indirect	No Air	#2	51.2	80.0	62.2	78.0	78.6	78.3	74.6	73.0	3.25
500 w, Bowl Silver Indirect	60	#2	34.7	54.1	69.9	78.0	79.3	77.8	75.4	74.8	4.29
500 w, Bare	No Air	#1	46.4	72.5	61.6	78.0	79.4	78.0	71.6	70.1	2.83
500 w, Bare	60	#1	26.3	41.0	69.5	78.0	81.0	78.3	75.3	74.3	3.10
500 w, Bare	No Air	#2	48.2	75.1	61.1	78.0	79.5	77.4	72.4	71.1	2.86
500 w, Bare	60	#2	33.0	51.4	68.5	78.0	80.4	77.6	74.2	73.9	3.47
300 w, Bowl Silver Indirect	No Air	#1	28.6	74.6	68.6	78.0	78.7	78.0	75.3	74.6	3.04
300 w, Bare	No Air	#1	27.2	71.0	67.8	78.0	79.2	77.7	74.0	73.8	2.67
6 Single-Tube Direct Fluorescent	No Air	#1	41.3	92.0	61.7	78.0	78.3	79.0	72.1	70.9	2.54
6 Single-Tube Direct Fluorescent	60	#1	18.1	40.4	71.1	78.0	79.2	78.5	75.1	75.0	2.62
2 Four-Tube Semi-Direct Fluorescent	No Air	#1	40.7	83.5	63.9	78.0	76.8	79.8	74.2	72.3	2.89
2 Four-Tube Semi-Direct Fluorescent	60	#1	19.8	40.6	71.9	78.0	78.6	79.5	75.8	75.5	3.24

[a] #1 Zinc oxide flat white paint. #2 Special copper phosphate heat absorbing paint.
[b] Measured by water flow and temperature rise.
[c] By surface couples.
[d] Aspirated couple.
[e] Corrected for Radiation.

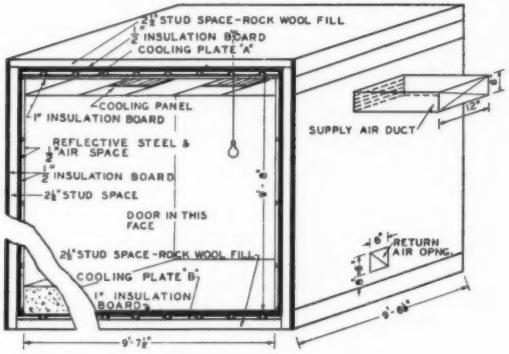

FIG. A-1. CONSTRUCTION OF TEST ROOM FOR LUMINAIRES

As in previous experiments, two types of ceiling paint were used, a commerical flat white paint containing zinc oxide and a special paint containing a small amount of copper phosphate.[5] Test data are presented in Table A-1.

The unit area performance of a cooling panel which comprises only a portion of a suspended ceiling is usually greater than the unit area performance of a panel in a continuously cooled ceiling, due in part to the effectiveness of the top of the panel

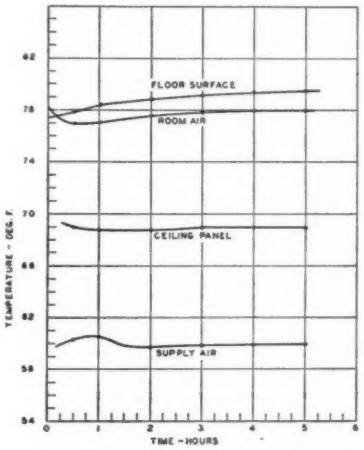

FIG. A-2. TEST DATA FOR THE NORMALLY LOADED ROOM, UNSTEADY STATE

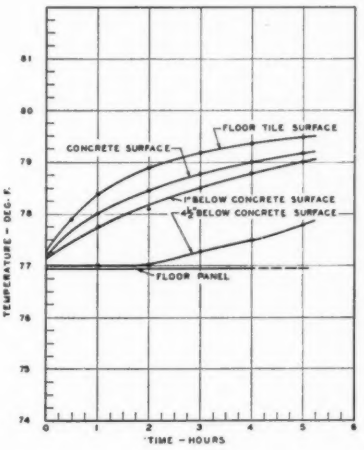

FIG. A-3. TEST DATA, NORMALLY LOADED ROOM, UNSTEADY STATE, TEMPERATURE WITHIN THE STRUCTURE

as a useful heat sink and, to some extent, to conduction from the adjacent metal acoustic pans.

There are combinations of air supply, type of luminaire, and ceiling paint which will cause a continuous ceiling to be more effective per unit area than a partially paneled ceiling.

The tables include a column for the fictitious value of surface conduction h. This factor is presented as a convenience only. As previously pointed out [1] it is not correct

TABLE A-2—SUMMARY, PART CEILING—UNSTEADY STATE

(Air Supply Where Used, 37.4 cfm—Floor Area, 92 sq ft)

Test No.	Lamp Arrangement	Air Supply F	Paint Used on Panels[a]	Panel Transfer[b,c]		Panel Temp[c,e] F
				Btu/ (Sq Ft) (Hr)	% of Lamp Load	
1	200 w Up — 300 w Down.....	60.0	#2	29.2	45.5	69.0
2	300 w Up — 500 w Down.....	60.0	#2	40.5	39.4	70.5
3	100 w Up — 200 w Down.....	60.5	#2	21.6	56.5	69.0
4	200 w Up — 300 w Down.....	70.0	#2	35.0	54.5	69.5

Test No.	Air Temp—5' Level[d]			Floor Surface Temp[c,e] F	Air Temp 4 in. Above Panel Center[e] F	Structure Temp Above Panel Center[e] F	% Storage of Lamp Load At End of		Apparent[f] "h" For Panel[e] Btu/ (Sq Ft) (Hr) (F Deg)
	Initial	3rd Hr	5th Hr				3rd Hr	5th Hr	
1	78.0	77.8	78.0	79.5	74.5	74.3	15.0	14.9	3.25
2	78.0	81.9	82.4	82.7	75.4	75.1	18.1	19.1	3.40
3	78.0	76.2	76.2	77.8	73.8	73.5	2.8	5.0	3.00
4	78.0	80.0	80.5	80.6	74.8	72.6	21.3	27.9	3.18

[a]#2, Special copper phosphate heat absorbing paint.
[b]Measured by water flow and temperature rise.
[c]By surface couples.
[d]Aspirated couple.
[e]Fifth hour value.
[f]For conditions of test only.

to present these results in the form of overall surface coefficients to cover both radiation and convection since they cannot be multiplied by a temperature difference as, for example, panel to room air, to predict results under any other conditions than those of this particular test.

The column on Independent Radiant Transfer shown in the previous experiments is omitted here as tests were not conducted for the separate measurement of this quantity.

Part Ceiling — Unsteady State

Tests were performed to check the effect of thermal storage with a panel cooling system for the load due to luminaries. In the test cubicle with a part panel ceiling, the 1 in. insulation board covering the floor panel was removed and 10 in. cinder concrete slab was poured directly on the panel (lower left corner Fig. A-1) in the following mixes:

5-½ in. of 1:2:5
3-½ in. of 1:3:6
1 in. of 1:2

¼-in. thick dark red asphalt tile was placed on top of the concrete. Thermocouples were inserted in the concrete slab. The original hung lighting fixtures were replaced by two prismatic glass reflectors; one to direct light to the ceiling panel, the other to the floor. This arrangement was used to minimize the absorption effect of radiant energy by the insulated and reflective walls.

The water temperature to the floor panel beneath the concrete was maintained constant at its initial value before the start of the test. The full panel ceiling water temperature was regulated throughout the test according to the surface temperature of the floor. The part panel temperature was set at an initial depression below the

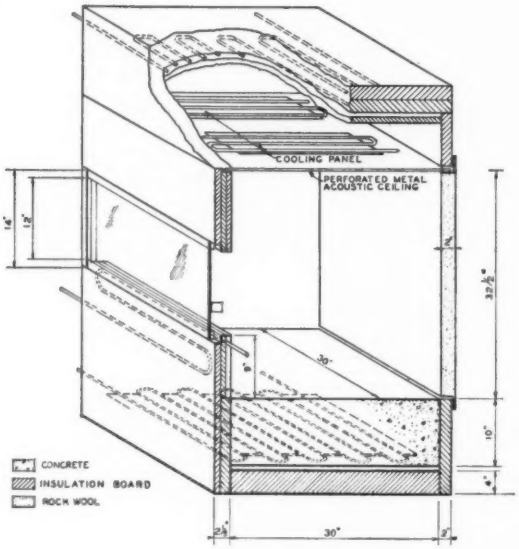

FIG. A-4. CONSTRUCTION OF MODEL FOR SOLAR TEST

room air temperature and maintained constant throughout the test. The air supply to the room was 0.4 cfm/sq ft and maintained at a constant temperature.

A lighting load of 500 watts was selected as the standard loading (5.4 watts per square foot) and, with the required panel depression to maintain the room air constant at its initial value with this loading, additional tests were conducted with the room 160 percent (800 watts) and 60 percent (300 watts) loaded. The effect of using a higher supply air temperature with the normal loading was checked by one test. All tests were conducted with heat-absorbing paint on the ceiling. Tests are summarized in Table A-2. Figs. A-2 and A-3 show the variation of structure and air temperatures with time for the normally loaded room.

In Test No. 1 for the normally loaded room, the panel temperature was initially established and continuously held at a value which would produce a 78 F room air temperature at the end of five hours. For Tests Nos. 2 and 3 this same panel temperature was maintained in order to check the deviations of room air and structure with

TABLE A-3—SOLAR MODEL PART PANEL CEILING

(Data for end of 3rd Hour)

Tests[a]	Single Clear Glass Inside White Ven Blind	Heat Absorb'g Glass Outside, Clear Glass Inside, White Ven Blind	Double Clear Glass, White Ven Blind Between Glass Panes	Double Clear Glass, ½ Drawn Inside White Ven Blind	Double Clear Glass, White Shade Between Glass Panes	Single Clear Glass Inside White Ven Blind	Single Clear Glass Inside White Ven Blind
	A	B	C	D	E	F	G
1. Ceiling Paint	Regular	Regular	Regular	Regular	Regular	Heat Absorb'g	Heat Absorb'g
2. Room Air Temperature (aspirated) F	78.0	78.0	78.0	78.0	78.0	78.0	78.0
3. Average Ceiling Panel Surface Temperature F	65.5	69.0	68.2	68.1	70.5	67.9	72.3
4. Average Sill Panel Surface Temperature F	66.1	69.2	68.4	68.0	71.0	68.1	72.5
5. Supply Air Temperature F	57.2	61.6	62.1	58.9	63.9	59.9	61.9
6. Return Air Temperature F	76.6	77.3	77.7	77.5	77.8	76.4	76.9
7. Floor Surface Temperature F	75.6	78.1	78.9	77.3	78.9	74.4	76.3
8. Rear Wall Surface Temperature F	79.2	78.9	80.0	79.9	80.9	79.0	78.9
9. Alzac Surface Temperature F	78.7	78.3	79.3	78.7	78.5	78.0	77.7
10. Wall Surface Temperature Above Window F	85.7	86.1	81.8	84.5	86.5	85.4	86.9
11. Shield Thermocouple (room air) F	78.3	78.2	78.2	78.7	78.1	78.4	78.1
12. Inside Shade or Venetian Temperature F	89.7	84.3	127.0	94.5	113.9	92.4	89.3
13. Inside Glass Temperature F	114.4	114.1	103.4	97.0	95.4	—	—
14. Outside Glass Temperature F	114.7	134.8	131.0	104.5	113.9	101.4	101.1
15. Front Ambient Temperature F	79.7	83.8	89.0	83.5	85.9	78.4	84.6
16. Ceiling Panel Transfer by Test Btu/hr	102.0	76.0	80.8	93.0	72.0	107.0	58.0
17. Sill Panel Transfer by Test Btu/hr	47.8	36.3	44.0	60.8	40.2	44.0	28.6
18. Removal by Air Supply Btu/hr	65.5	53.1	53.0	63.0	47.0	55.8	50.7
19. Transfer Through Walls[b] Btu/hr	− 42.1	− 8.7	26.4	− 30.8	− 4.8	− 33.0	− 28.7
20. Storage (average)[c] Btu/hr	5.0	− 9.0	− 6.0	35.0	1.0	9.0	9.0
21. Total Room Load[d] Btu/hr	262.4	165.1	145.4	282.6	165.0	248.8	175.0
22. Normal Incident (meter) Btu/hr	536	463	509	515	498	490	343
23. Reflection, % of Normal Incident (meter)	35	12	38	—	42	36	37
24. Solar Altitude Deg	35	35	35	35	35	35	60
25. Solar Azimuth Deg	0	0	0	0	0	0	0
26. Room Load from Test, % of Normal Incident	49.0	35.6	28.6	55.0	33.1	50.8	51.0
27. Apparent "h" for Ceiling Panel Btu/(sq ft) (hr) (F deg)	3.26	3.38	3.30	3.75	3.83	4.22	4.07
28. Apparent "h" for Sill Panel Btu/(sq ft) (hr) (F deg)	2.14	2.19	2.43	3.22	3.05	2.36	2.76
29. Calc Room Load, % of Normal Incident 78 F Ambient (5 mph Wind)	46.7	30.2	24.8	51.9	30.0	48.8	46.6
30. Calc Room Load, % of Normal Incident 93 F Ambient (5 mph Wind)	50.6	33.1	27.5	57.5	33.0	52.8	52.7

[a] Window fully covered where shading devices used except where noted, venetian blind slats set at 55 deg.
[b] Positive values — transfer into room.
[c] Positive values — increased storage.
[d] Values of line 21 = values of lines [(16 + 17 + 18 + 20) — 19].
Ceiling panel area — 2.80 sq ft. Sill panel area — 1.88 sq ft. Floor area — 6.25 sq ft.

time under different conditions of load. This variation in temperature of 6.2 F compares with the approximately 4.3 F deg variation for a continuously cooled ceiling, as shown in Fig. B-7.

In the overloaded room, with a panel ceiling, the rise in structure temperature is less than for a conventional air system. Floor, ceiling, and mean radiant temperatures are lowered.

Test No. 4 is for the same conditions as Test No. 1 but with the air introduced at 70 F instead of 60 F.

COOLING PANEL PERFORMANCE WITH LOAD DUE TO SOLAR RADIATION

Part Ceiling

The original experiments [3] are here extended to include data on a suspended ceiling consisting, in part, of panels. Data were obtained for additional types of shading device and angles of incidence other than 35 deg.

The solar model previously described was modified by the addition of a suspended ceiling of which 40 percent was in the form of aluminum panels and 60 percent of steel perforated plate with acoustic pads above, as shown in Fig. A-4. The panels were 20 in. long by 9 in. wide and separated by 6 in. of perforated steel ceiling. The original continuous panel ceiling was raised 9 in. and covered with 1 in. of insulating board on the under side to represent the thermal resistance of a concrete slab of the floor above in a multi-story building. The openings created by the raising of this panel were sealed with insulating board to conform with the original walls of the model. The water piping arrangement remained as previously described. Additional piping was installed for the new ceiling. For these tests the continuous ceiling panel temperature was adjusted to be the same as the surface temperature of the floor in order to simulate the steady state condition. Other than for this point, the procedure was the same as in the original test with a continuous ceiling [3].

The test data are presented in Table A-3. The performance per unit area, where the panel represents 40 percent of the ceiling, is better than the performance for the continuously cooled ceiling, as previously presented [3]. Lines 27 and 28 have been included for convenient comparison. Line 20, heat storage within the floor, is small with the exception of the one test with a half-drawn blind, in which case the radiant energy from the artificial sun was permitted to strike the floor.

Columns B, C and E indicate that the scheme of placing a venetian blind or white shade between two sash is approximately equivalent to using heat absorbing glass outside, clear glass inside, plus a white venetian blind. Both methods effect a significant decrease in the solar heat load.

APPENDIX B

SPACE COOLING SYSTEMS ANALYZED WITH A HYDRAULIC ANALOGUE

In a previous paper[2] the difference between the real behavior of a conventional air cooling system and the design concept is indicated by means of a hydraulic analogue. These studies have been continued with the analogue modified as required.

Fig. B-1 indicates the performance of a conventional air cooling system for the interior zone of a multi-story building, with eleven hours of operation of the system and nine hours full load due to the lighting system. The illumination in this case has been assumed as by direct filament lamps and a load equivalent of 2.7 watts per square foot. At the end of the third day the actual cooling requirements are approximately 83 percent of that calculated for the removal of the internal sensible load.

Fig. B-2 approximates the performance of a conventional system, with 3.8 watts

c Positive values — increased storage.
d Values of line 21 = values of lines [(10 + 17 + 18 + 20) — 19].
Ceiling panel area — 2.80 sq ft. Sill panel area — 1.88 sq ft. Floor area — 6.35 sq ft.

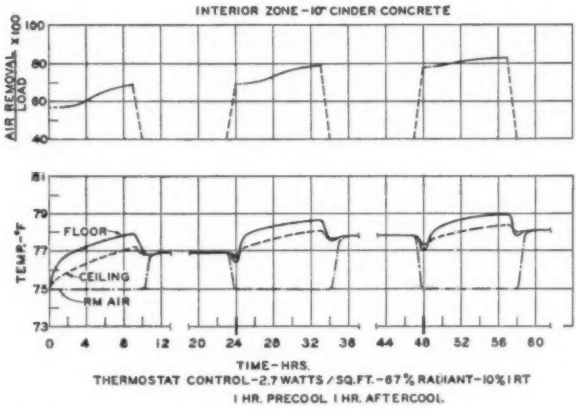

FIG. B-1. PERFORMANCE OF AN AIR COOLING SYSTEM WITH DIRECT LIGHTING BY FILAMENT LAMPS

per square foot, assuming a luminaire having a lesser radiant component than a filament lamp, or a rough approximation of a load consisting of filament lamps and people. In this experiment the room air has been assumed to be permitted to vary from 73.6 F to 76.8 F. The actual sensible cooling requirements in the afternoon are indicated as less than in the morning and this is a fair indication of the performance of a system which is under capacity as to means of sensible heat removal.

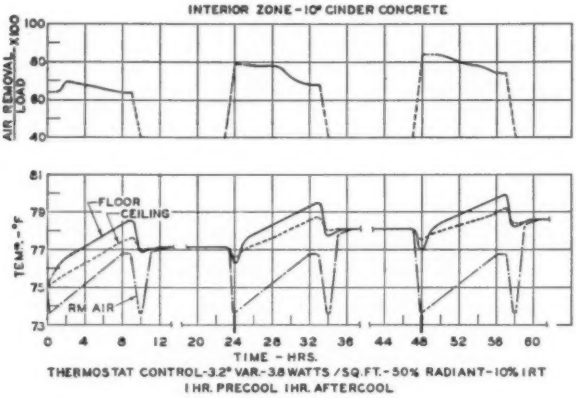

FIG. B-2. PERFORMANCE OF AN AIR COOLING SYSTEM ASSUMING A LOWER PERCENTAGE OF HEAT GAIN IN THE FORM OF HIGH TEMPERATURE RADIATION AS WOULD BE APPROXIMATED BY DIRECT FILAMENT LIGHTING AND PEOPLE

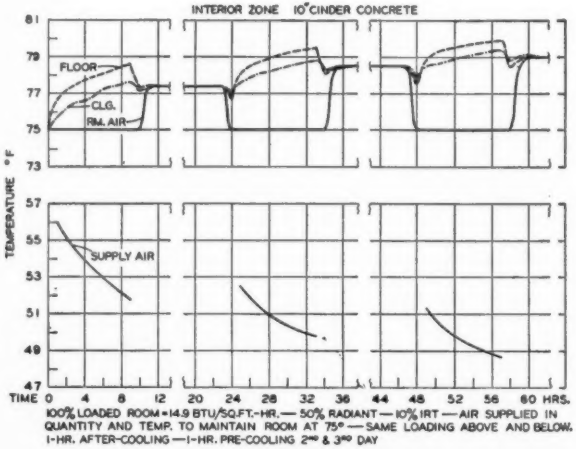
INTERIOR ZONE 10" CINDER CONCRETE
TEMPERATURE °F
FLOOR
CLG.
RM. AIR
SUPPLY AIR
TIME 0 4 8 12 20 24 28 32 36 44 48 52 56 60 HRS.
100% LOADED ROOM = 14.9 BTU/SQ.FT.-HR. — 50% RADIANT — 10% IRT — AIR SUPPLIED IN QUANTITY AND TEMP. TO MAINTAIN ROOM AT 75° — SAME LOADING ABOVE AND BELOW. 1-HR. AFTER-COOLING — 1-HR. PRE-COOLING 2ND & 3RD DAY

FIG. B-3. AIR COOLING SYSTEMS—REQUIRED SUPPLY AIR TEMPERATURE *vs.* TIME TO MAINTAIN CONSTANT TEMPERATURE

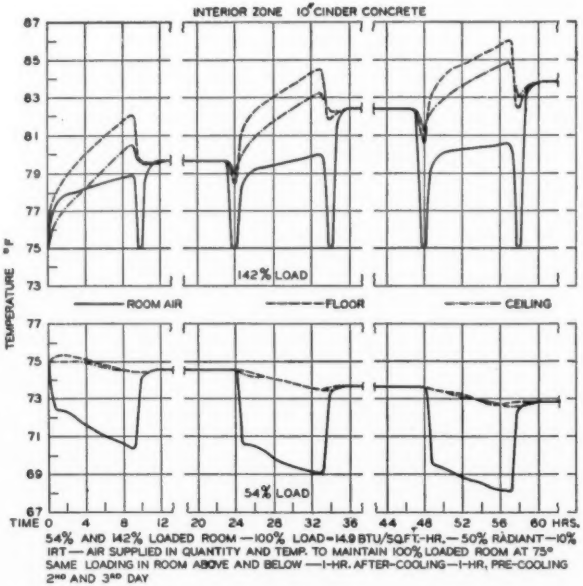
INTERIOR ZONE 10" CINDER CONCRETE
TEMPERATURE °F
142% LOAD
ROOM AIR
FLOOR
CEILING
54% LOAD
TIME 0 4 8 12 20 24 28 32 36 44 48 52 56 60 HRS.
54% AND 142% LOADED ROOM — 100% LOAD = 14.9 BTU/SQ.FT.-HR. — 50% RADIANT — 10% IRT — AIR SUPPLIED IN QUANTITY AND TEMP. TO MAINTAIN 100% LOADED ROOM AT 75° SAME LOADING IN ROOM ABOVE AND BELOW — 1-HR. AFTER-COOLING — 1-HR. PRE-COOLING 2ND AND 3RD DAY

FIG. B-4. AIR COOLING SYSTEMS—TEMPERATURE FOR A STRUCTURE AND AIR FOR A 54 PERCENT AND A 142 PERCENT LOADED ROOM WITH AIR SUPPLY FOR A NORMALLY LOADED ROOM

In order to establish some boundaries to the problem of the need for individual control for differently loaded rooms on the same zone of a conventional system, the problem was set up on the Analogue as follows:

In Fig. B-3 the supply air temperature to maintain constant room temperature was determined for a three-day period. At the end of the third day, the required air temperature was approximately 49 F to maintain 75 F, and this difference of 26 deg

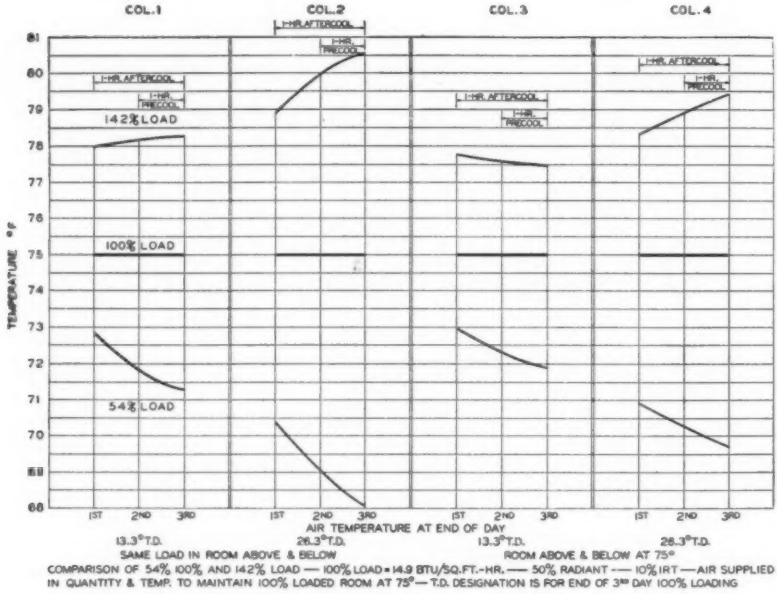

FIG. B-5. AIR COOLING SYSTEMS—RESULTS OF TESTS OF REQUIRED ROOM AIR TEMPERATURE AT THE END OF DAY WHEN SUPPLY AIR IS INTRODUCED TO MAINTAIN THE 100 PERCENT LOADED ROOM AIR TEMPERATURE AT 75 F DURING EACH DAY

was assumed as establishing the design temperature difference between room air and air introduced for cooling. As shown in Fig. B-1, the full cooling effect of design air is not required at the end of the third day so that this temperature depression of supply air would correspond to a slightly greater depression in conventional design. The air temperature schedule diagrammed in Fig. B-3 was then applied on the Analogue to a room which had 142 percent of full load and to a room having 54 percent of full load. The temperature results of structure and air are diagrammed in Fig. B-4.

For the result in Fig. B-4 it was assumed that the space directly above and directly below were subject to the same overload and the same underload as the area stated. The experiment on the Analogue was repeated but modified in the assumption that the rooms directly above and directly below were maintained at normal load. This condition was simulated by maintaining a known schedule of mid-point temperatures,

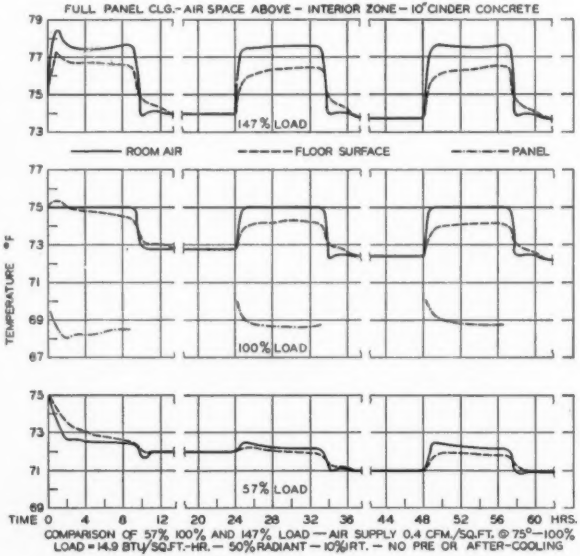
FULL PANEL CLG.-AIR SPACE ABOVE - INTERIOR ZONE - 10" CINDER CONCRETE
147% LOAD
ROOM AIR
FLOOR SURFACE
PANEL
100% LOAD
57% LOAD
TEMPERATURE °F
TIME 0 4 8 12 20 24 28 32 36 44 48 52 56 60 HRS.
COMPARISON OF 57% 100% AND 147% LOAD — AIR SUPPLY 0.4 CFM./SQ.FT. @ 75°—100% LOAD = 14.9 BTU/SQ.FT.-HR. — 50% RADIANT — 10% IRT. — NO PRE OR AFTER-COOLING

FIG. B-6. ROOM AIR AND FLOOR SURFACE TEMPERATURES FOR A PANEL COOLING SYSTEM IN WHICH THE PANEL IS THE ONLY MEANS OF COOLING. THE PANEL TEMPERATURE REQUIRED TO MAINTAIN THE 100 PERCENT LOADED ROOM AIR TEMPERATURE AT 75 F IS USED IN THE OVERLOADED AND UNDERLOADED ROOMS

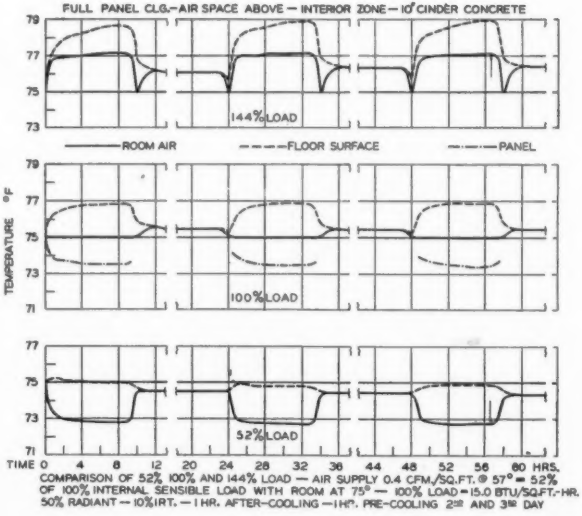
FULL PANEL CLG.-AIR SPACE ABOVE — INTERIOR ZONE — 10" CINDER CONCRETE
144% LOAD
ROOM AIR
FLOOR SURFACE
PANEL
100% LOAD
52% LOAD
TEMPERATURE °F
TIME 0 4 8 12 20 24 28 32 36 44 48 52 56 60 HRS.
COMPARISON OF 52% 100% AND 144% LOAD — AIR SUPPLY 0.4 CFM./SQ.FT. @ 57° = 52% OF 100% INTERNAL SENSIBLE LOAD WITH ROOM AT 75° — 100% LOAD = 15.0 BTU/SQ.FT.-HR. 50% RADIANT — 10% IRT. — 1 HR. AFTER-COOLING — 1 HR. PRE-COOLING 2ND AND 3RD DAY

FIG. B-7. RESULTANT ROOM AIR AND FLOOR SURFACE TEMPERATURES FOR A PANEL COOLING SYSTEM IN WHICH BOTH SUPPLY AIR AND PANEL PROVIDE ROOM COOLING

floor and ceiling slab, as previously determined as an average for a run with a normally loaded room and a run for an overloaded or underloaded room. This is not an absolute parallel but the assumption is within the accuracy of the experiment.

The results of the two series of experiments are shown in Fig. B-5, for two depressions of air supply below room temperature as measured at the end of the third day.

Comparison of Columns 1 and 3, 4 and 5, shows the importance of transmission through the floor and ceiling slabs. Transmission through partition walls would be expected to have a similar effect.

The Analogue was modified to simulate a continuous suspended panel ceiling. Briefly, this is accomplished as follows:

The Independent Radiant Transfer is independent of the panel temperature. The effect of the Independent Radiant Transfer could, therefore, be simulated by not introducing into the system the quantity of fluid corresponding to this energy. A single tube, corresponding to the thermal capacity of an aluminum ceiling with coolant tubes, was used to simulate the panel. This tube was then connected to suitable conductors representing radiation to the floor and ceiling above and other conductors representing convection to the air below and above the panel. The temperature of the panel was simulated by a valve connection which controlled the height of liquid in the tube to simulate the panel temperature.

The Analogue was operated to establish the required panel temperature for the normally loaded room, with a selected air supply of 0.4 cfm at room temperature, as shown in the middle diagram of Fig. B-6. This schedule of panel temperatures was then applied to the overloaded and underloaded room, as shown in the upper and lower diagrams respectively. The experiment was repeated with the modification that the ventilating air was assumed as introduced 18 deg below the temperature of the normally loaded room, with the results as shown in Fig. B-7. Where the full ceiling is used in this latter case, the required panel depression below room air temperature is less than 2 deg.

DISCUSSION

F. W. HUTCHINSON, Berkeley, Calif.: In my opinion, the idea behind this paper is the greatest in applied radiation for the last 10 years.

EDWARD SIMONS, San Francisco, Calif.: It has been my pleasure to follow Mr. Leopold's development since 1947. Those wishing to study the subject will want to follow the papers mentioned in the Bibliography.

Since 1947, Mr. Leopold has gone into a very thorough study of conductance, with respect to both convection, radiation and conduction. He separated them into wall, ceiling and panel surfaces, as he had to establish values for each. Subsequently, he adopted the term IRT, which was the independent radiant transfer to the ceiling panel, exclusive of that which was absorbed by other parts of the room. It was an important breakdown. He also began to develop the conductances to these surfaces, later applied to his analogue.

I would suggest, however, that the Appendix be extended to include the diagram of the analogue, and also that the tube representing the panel and its temperature and the conductances into the ceiling be shown. This will be useful information for those who want to follow the explanation. This is another important paper on the subject; and Figs. B-3, B-4 and B-7 are valuable in showing comparisons in a room supplied with convection cooling as compared to a room supplied with air at a fixed temperature and quantity, and using fixed panel temperature.

If we may take a certain level of mean radiant temperature *vs.* ambient temperature as a measure of comfort, setting that as a comfort equation, and then assume the

average surface temperature as being equivalent to the mean radiant for our particular discussion, we will arrive at these findings: In Fig. B-7, for cooling panel (a normally loaded room) the average temperature of the ceiling and floor, plus that of the air, equals about 150 deg. In the normally loaded room, Fig. B-3, under the conduction and convection cooling (the normal type), it is found that the average temperature of the air is 75 F, but this rises to about 79½ as the average temperature of ceiling and floor rises. This is a departure from that basis of the set figure of 4½ deg. If we take the overloaded rooms with the panels, Fig. B-7, it is found that this type of departure, with fixed air supply and fixed panel, is only about 3.1 F, a lesser departure than for the normal type of room. The underloaded panel room has only a depression below that of normal of about 3.1 F. However, on the overloaded room with air cooling only, the result is a departure of 16 deg in the plus direction, and nine in the minus direction on the underloaded room.

Therefore, with this type of a system, and with our analysis as the basis of this analogue and thinking, it is seen that a definite approach is being made to a steady energy device with which we can begin to apply general energy equations regarding what enters and what emerges.

G. LORNE WIGGS, Montreal, Que., Canada: I would like to congratulate Mr. Leopold on his excellent paper. It should be thought-provoking for many of our members.

The author has referred to this system as a panel and air cooling system. This is probably better than simply a radiant cooling system. On the other hand, if the term *radiant heating system* can be applied to a system heating predominantly by radiation, then, probably, the expression *radiant cooling system* is a satisfactory name for this type of cooling. In England, such a system would be referred to as a panel warming and cooling system, with moderate air conditioning.

I would like to point out that panel cooling is not new, as you might think. It was conceived and developed by Richard Crittall & Co., Ltd., England. Apart from the experimental installations they made, it was first installed by them in the Imperial Chemical House, London, in 1935. That system has, I think, about 900 tons of refrigeration to cool the water and air, giving you an idea of the size of the system. It has also been installed in many other buildings in Europe, but it has not been used here on this continent, except for the installation mentioned by Mr. Leopold.

As far as I can determine, there is no real scientific basis for the designs that were made in England and Europe. Consequently, this paper is probably the first publication that would permit design with a little more confidence than is possible with the rules of thumb used in Europe.

To give you some idea of this, the Manufacturers Life Bldg. installation will contain about 240,000 sq ft of floor area. It is located in a climate with a maximum temperature reaching almost as high as that of Philadelphia, but whose duration of high summer temperature is not nearly as long. Winter temperatures there, on occasions, drop to about 25 deg below zero. As a result, air conditioning systems must be capable of operating satisfactorily over a wide range in climatic conditions.

In the Manufacturers Life Bldg., on a typical floor, about 12 percent of the ceiling area is taken by the peripheral panels; 1 percent of the ceiling is occupied by the diffuser; about 22 percent by the luminaires; and the remaining 65 percent consists of perforated steel acoustic panels which, with the high noise reduction coefficient of those panels, will give a satisfactory acoustic treatment.

In the Canadian installation, Canadian-made extrusions are used. These differ in some details from those described by the author of this paper. They are actually 0.080 in. thick. Admittedly, this requires more aluminum, but the difficulty of extruding them is less, so that the manufacturers are delivering at a somewhat lower price than that at which they would have provided the thinner panels.

In Norway they have been developing and using perforated aluminum panels of sheet aluminum rather than extrusions. The entire ceiling is covered with aluminum

panels fastened to pipes through heat-conducting clips. Warm water is circulated through the pipes in winter, and chilled water in the summer. Thus, heating or cooling is obtained as required. The pipes themselves serve as supports for the panels. It is probably true that this system may be less expensive than the extruded panel type in many cases.

In closing, I would just like to say that this system shows great potential, particularly in such areas as operating rooms, where drafts are sometimes serious. In addition, with television cameras being used in operating rooms for instruction purposes, methods for removing heat, without increasing air quantities, can be tremendously important. The system also can be used to great advantage in textile mills and other places where there are large concentrated loads, as well as considerable noise.

A. T. JORN, Libertyville, Ill.: This is a fine paper in that it has covered the radiant panel for cooling in regard to its limitations, its attributes, its physical constructions, its integration with other functional units, and its probable relationship to conventional air conditioning. Industry, and in particular research men, appreciate consulting engineers who are enthusiastic about new tools for providing human comfort, and who investigate for themselves the best means for applying research developments and inventions. Such enthusiasm expedites the moving of new concepts from the laboratory and field-trial stage into widespread public use.

This is also one of the first technical papers to call wide attention to the new radiant metal panels and to present some of the research work which has been quietly carried out with them in various points about the country. This paper gives such a broad perspective to anyone interested in the application of metal radiant panels for cooling, that I would like to add some information on work done in my particular research and development group.

Mr. Leopold has pointed out several ways of applying cooling panels: (1) as panels, essentially at room temperature, grouped around windows to absorb solar radiation; (2) as panels (or luminaires), essentially at room temperature, strategically placed about the ceiling area to absorb the heat emitted by the illumination units; and (3) as panels, essentially below room temperature, installed to cover the entire ceiling area for removing all or any apportioned part of the sensible heat caused by solar radiation, artificial illumination, electrical equipment, and human occupancy.

The third application of the panels was not discussed at length by the author. Its limitations were due to (1) relatively high cost of cooling panels; (2) mutually eliminating characteristics of acoustic panels and radiant panels; (3) problems involving symmetry of appearances of the completed ceiling; and (4) possible condensation problems if panel temperatures were lowered sufficiently to be useful in absorbing heat other than high temperature radiation.

It is upon this third method of using cooling panels and the associated problems that I should like to comment:

1. Most types of metal radiant panels as mentioned in the paper tend to be expensive, since the coolant carrying coil is attached to the radiating surface in an expensive manner, either by brazing or by expansion into a pre-formed groove. The metal surface is composed of relatively thick material according to the manufacturing processes; the coil must be of expensive material also due to manufacturing processes; and installation of the finished panel is a relatively slow and awkward process.

The cost consideration can be reduced appreciably by a design in which the radiating surface can be rapidly attached to the coil after installation by means of integral clamps which make a good and uniform thermal contact. The panel is made of a minimum thickness of material consistent with structural strength (about $\frac{1}{5}$ to $\frac{1}{2}$ that required in some metal radiant panels). The panel material is that which is most economical for the required thermal capacity. Steel and copper as well as aluminum of maximum thermal conductivity, have economic application. The coil may be of any standard coil material.

2. Conflict between radiant and acoustical panels can be eliminated by perforating the panels, as pointed out in the paper. The design just mentioned would not interfere with the all-over, approximately 10 percent open, perforation pattern generally used on acoustic ceilings. The only purpose of the perforations in metal pan acoustic ceilings is to allow free flow of the sound waves out of the room and into contact with an absorber. In the suggested design, a single sound absorbing and insulating blanket was used above the coil to obtain thermal insulation, acoustic absorption, and vapor resistance. Such economy of material should lessen the acoustic and radiant panel price difference.

3. A cooled luminaire has been developed to be used with this suggested design. A single modular-patterned unit combines the functions of acoustics and radiation, and air conditioning outlets are available having the same surface pattern.

4. As Mr. Leopold has shown in Appendix B, it is theoretically possible, according to this Hydraulic Analogue, to depress the temperature of a radiant panel covering practically the entire ceiling area, and, without going too close to the dewpoint, to absorb not only radiation from high temperature sources such as the sun and the lights, but also heat from room surfaces, furnishings, occupants, and air. We knew this to be not only theoretically, but also practically, possible because it has actually been accomplished in our drafting room and in the office of our company's president.

I believe there is a basic advantage in covering the entire ceiling area with cooled panels and luminaires. My belief is based upon the sense of comfort which can be *felt* when cooling panels are employed in this manner. Walking from an adjoining room of identical construction and exposure into the drafting room, previously mentioned, often led to the sensation of a 10 to 15 deg cooler temperature, even though the drafting room had no cooling facilities other than the radiant ceiling, and even though the air temperature in the two rooms was identical.

The only explanations I can give you of this phenomenon are: (1) the furnishings, walls, and floor of the room were, due to the effect of the ceiling, cooler to the touch than those in the adjoining room, so one's body would tend to lose heat to them more readily; (2) in a normal room in the summer, air warmed by lights, sun, people, etc., rises to the ceiling and warms that surface to temperatures approaching those at which we normally design and operate radiant heating panels for our winter comfort. Perhaps, then, the most notable service performed in covering the entire ceiling surface with a cooling panel is that of eliminating this normal summer condition of radiant heat emission from the ceiling.

To fully comprehend the possible implications of the latter concept, it must be kept in mind that the heat in warm air which has risen to the ceiling is essentially removed from our comfort consideration if it cannot warm a surface. This is true because it is impossible for air to radiate energy; only a solid object warmed by the air can radiate the heat energy to the living zone where it will affect comfort.

If this concept is true, it would explain why it is so desirable to employ a panel covering the entire ceiling surface rather than to use small dispersed panel areas. If the panel covers the entire ceiling, no surface is heated by the rising warm air currents. (Upper walls are apparently kept sufficiently cool by direct radiation to the cooled ceiling surface).

I mention this unproved explanation only because the data on the actual heat removed by these room-size ceiling panels affords no explanation of the relative sense of comfort *felt* upon entering the rooms.

Incidentally, during the winter, drafting room occupants were unaware of room air temperature depression, by ventilation, to as low as 63 F.

I should like to ask Mr. Leopold what Btu-per-sq ft heating capacities he has observed in either his commercial metal panel installations or in tests. I should be interested also in the percentage of increased cooling panel performance produced by the special copper phosphate heat absorbing paint he mentioned, though perhaps this information is to be found in the data section of the paper.

R. A. GONZALEZ, Dayton, Ohio: Thanks to Mr. Leopold for a very fine paper. It is interesting to read some of Mr. Leopold's comments about ¼ of a cfm per sq ft and ⅒ of a cfm per sq ft. Perhaps we have not investigated nearly as much as we might the matter of the air motion and the variants in tolerance or desirability of this air motion.

It is fairly well established that in the South, the Southwest and in Western areas we desire or require, in order to satisfy the average person, a greater air motion than is tolerated in the Eastern areas of the country. The designer often-times finds himself without enough basic coil air requirement to satisfy the local idea of air motion required. This is sometimes true of exterior zones and almost invariably the case on interior zones. Even the increased lighting load of late years has not changed this situation. That is an additional phase or factor of air conditioning that can be investigated.

Certainly, Mr. Leopold's approach has given us a very valuable tool to use in many cases where we cannot further reduce the load by improved installation or by reducing the wattage.

It should prove a valuable tool for use in designing systems for individual persons who are hyper-sensitive to air motion. The full advantages of air conditioning can be provided for these cases with the minimum of air motion.

AUTHOR'S CLOSURE: I deeply appreciate Professor Hutchinson's comment. In late 1945 or early 1946, when I first realized that a large portion of the energy of a luminaire would be absorbed by the structure and could be dissipated even by a panel maintained at, or above, room temperature, I telephoned Professor Hutchinson to try to arrange for a laboratory check. It was not possible to change his laboratory schedule, but he kindly agreed to run an over-night test in the research residence on the Purdue campus. Professor Hutchinson phoned the results the next morning and, fortunately, the tests confirmed the theoretical conclusions.

I appreciate the comment by Mr. Simons, vice-president of *The American Society of Refrigerating Engineers*, and the fact that he has travelled 3000 miles to discuss this paper.

I agree with Mr. Simons on the desirability of including the various analogue set-ups for Appendix B and must plead length of paper and lack of space as the reason for its omission.

In regard to the mean radiant temperature, I am in general agreement with Mr. Simons' statement, with the following qualification. For simplicity, I have analyzed the unit area in an infinite interior zone. If walls or furniture had been present, the storage would have been approximately the same, but the increase in mean radiant temperature would have been considerably less.

Mr. Wiggs has suggested that, since we call panel heating systems *radiant heating systems*, the term *radiant cooling systems*, rather than *panel cooling systems*, might be used. Since convection has such a large part both in heating and cooling, I think it is correct to refer to *panel heating and cooling systems*. The term *radiant heating* has been generally misunderstood by the public, since it was confused with heating by small radiant sources at high temperature, a system in which the convection effect is greatly minimized. F. W. Hutchinson has discussed this point in an article titled *The Myth of the "Indoor Sun" (Heating, Piping & Air Conditioning,* December 1949, p. 74).

The statement that extruded panels 0.08 in. thick have proved cheaper in Canada than the original 0.062 in. thick panels manufactured in the United States requires explanation. The panels in the United States are actually less costly than the Canadian panels, as the United States manufacturer had the know-how to extrude panels of the thinner material.

The use of panels for special areas of high heat load should be carefully studied. They will be most effective where a large percentage of the internal heat load is in

the form of radiant energy and where the temperature of the panel can be depressed without approaching the maintained room dewpoint.

Mr. Jorn quotes me as utilizing panels grouped around windows and on the ceiling at *essentially room temperature.* I have pointed out that the ceiling panels, in particular, will remove a considerable amount of high temperature radiant energy if maintained at, or even above room temperature. However, I did not state, and do not mean to imply, that these panels should be operated at room temperature. On the contrary, if the temperature of these panels is lowered, they remove additional energy in the form of low temperature radiation and by convection. In my paper, I have referred to temperatures substantially less than room level in describing the system and in all the tests. Mr. Jorn's speculation as to why I worked at room temperature also does not conform with the statements in the paper.

There are no *mutually eliminating characteristics of acoustic panels and radiant panels.* The radiant panels can be perforated for acoustic treatment if additional treatment were necessary without appreciable effect on their thermal properties. This was so stated. The discrete panel has been described in this paper because it was found unnecessary to treat a large part of the ceiling. This left the remaining portion of the ceiling construction to be selected according to the needs.

Mr. Jorn's ensuing discussion is largely concerned with the economics of cooling an entire ceiling with a thermal bond which is formed by forcing the ceiling material against the coolant pipes with a spring clip, as compared with the formal panel construction where there is a tight thermal bond. I have no brief for either method. This problem, I hope, will be solved eventually by competition.

Mr. Jorn refers to radiant cooling. I hope that the term panel cooling remains because it is less misleading to the public.

As Mr. Jorn implied, and as I have stated in my paper, I believe that there is a field for panel cooling without the introduction of air for dehumidification. I will caution, however, that such applications must be very carefully studied where condensation would cause damage.

I have noted the comment on the improved feeling of comfort in the panel cooled, as compared with the conventional room construction, and again, I recommend caution. In our tests in the cubicle described in the first paper of this series, there was a continuously cooled ceiling and, as far as spot judgment is concerned, the lowering of the mean radiant temperature had a minor effect and about as much as would be anticipated from previous work. The unusual observation, if correct, was the effect on winter comfort that I mentioned and which Mr. Jorn confirms. I do not know whether or not the comparison is valid because the ceiling panels in our installation were usually below room temperature and were never used for heating.

My observation does not check Mr. Jorn's speculation that there would be noticeable difference in thermal currents with a partly cooled ceiling, and that the continuously cooled ceiling, therefore, was preferable. Our tests indicated a markedly low air motion and an even temperature, floor to ceiling, to within a few inches of the panels. There were no noticeable comfort effects due to the relative position of the occupant's head and the pattern of panels on the ceiling.

Again, in the absence of carefully controlled tests, I should like to advise caution due to the difficulty of drawing any conclusions on comfort.

The question Mr. Gonzalez raised caused me some concern when I first embarked on the design of an actual panel cooling system. There were good indications in our test cubicle that the lack of air motion would not be noticeable, but there was still the problem of applying these to regular working quarters. I agree that there are an appreciable number of individuals in the southwest who require what is generally considered to be excessive air motion. Consequently, in a building in Texas, air conditioned in 1938, I made provision in the outlet design which was to provide a draft for those who wanted it. Originally, a number of the office occupants used this feature.

Less than 10 percent of the offices are now adjusted for anything but the most draftless operation.

In the Time, Inc. installation, which has operated through three summers, provision was made to increase the air velocity at the outlets, if required, by selecting oversized outlets and blocking a portion of them. It was found, however, that even with the small quantity of air being introduced through an over-sized outlet, the sensation of freshness, liveness, or however you will describe it, was acceptable and might be classified as superior.

In closing, I wish to thank all of the discussers for their considered opinions.

1410

No. 1410

THE EFFICIENCY OF BITUMINOUS-COAL-BURNING SPACE HEATERS†

By J. W. Tieman* and F. L. Bagby**, Columbus, Ohio

SEVERAL methods for rating coal-burning heaters have often been proposed, but none has the advantages of using a so-called calorimeter room designed specifically for this purpose. The true rating of space heaters requires a direct measurement of the output, obtainable in such a room, inasmuch as the magnitude of the unaccounted-for losses inherent to the burning of bituminous coal can only be determined by difference from a heat balance involving all known heat losses, the measured heat output, and the known heat input. The quantity of unaccounted-for heat required to complete the balance is, of course, the loss in unburned fuel escaping with the flue gases.

As a part of the program to evaluate improved heaters burning bituminous coal, Bituminous Coal Research, Inc., jointly with a group of stove manufacturers, authorized the construction of a test calorimeter room at *Battelle Memorial Institute* which is described later.

The object of the work discussed in this paper was to obtain the relation between heater efficiency and the rate of heat output, for the three general types of coal heaters currently being sold commercially; these are the conventional surface-fired type, the refractory-lined, (so-called magazine) deep-fire-pot type, and the crossfeed type smokeless stove. The scope of the work also included the determination of the added useful heat that is available within the house from the typical flue-pipe installation used between heater and chimney. Curves showing the overall house-heating efficiency that may be expected, when the heat gain from the flue pipe and from the chimney are taken into account, are presented for the heaters tested.

General Program of Tests

A total of four heaters, including two variations of the deep-fire-pot type, were tested in this program. The two deep-fire-pot heaters differed in the

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† Information presented in this paper was obtained from work sponsored at *Battelle Memorial Institute* by Bituminous Coal Research, Inc., the research agency of the bituminous coal industry.

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design of the inner refractory lining and were chosen as being representative of the several types now being manufactured.

The tests of these heaters and of the smokeless heater consisted of a series of burning cycles at different rates, but covering the normal range of operating rates of such units. The conventional surface-fired unit was tested at one rate only. Each test cycle for the first three heaters was run at a constant, preassigned flue-gas temperature which was varied from test to test. To maintain this temperature, the applied draft and air-inlet dampers were adjusted as often as necessary; furthermore, attention was given to the grates and to the fuel bed as often as required. The only limitation imposed was that of not exceeding a maximum stack draft of 0.06 in. of water during any test.

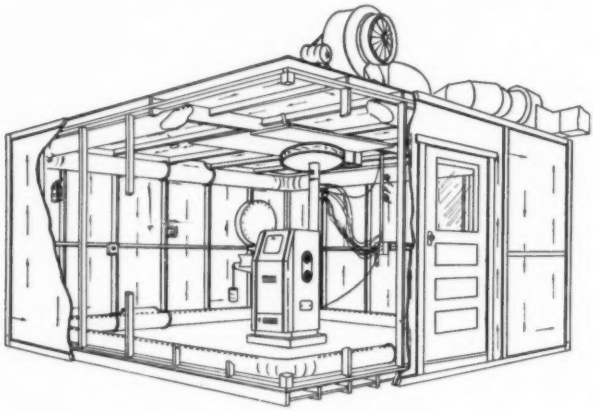

FIG. 1. CUTAWAY VIEW OF CALORIMETER ROOM

Tests conducted in this manner, while deviating from the accepted procedure for output rating tests, permitted operation of the heaters on a basis which insured reproducibility of results, since flue-gas temperature is the variable in heater tests that can be most readily controlled.

Data presented for a conventional surface-fired heater in this paper were obtained from rating tests run several years prior to those conducted on the other three heaters. For that reason, the imposed limitations differed in that the flue-gas temperature was not held at a constant value, but, instead, was allowed to vary below a maximum of 900 F and there was no grate shaking or poking of the fuel bed; the same limitation was used on maximum draft. Performance curves plotted from the data obtained with this heater during four-hour tests were felt adequate for providing a reasonably accurate basis for comparison with the results of tests with the other heaters.

CALORIMETER ROOM

Fig. 1 is a cutaway view of the calorimeter room in which these tests were conducted. The room is not a calorimeter room in the sense of adiabatic

calorimeters, but was designed to permit determination of heat flow through the walls which were of light enough construction not to offer large heat capacity, thus minimizing fly-wheel effects with change in output.

Sheets of ½-in. insulating board are attached to the simple stud and joist framework shown, which gives a floor area of 12 × 12 ft square and a ceiling height of 8 ft. Differential thermocouples are provided to indicate the average temperature difference between the inside and outside of the wall and ceiling boards. Two blowers are mounted on the roof to force air into and withdraw air from the room through suitable ducts. Dampers at the blower inlets are provided to maintain a room pressure very close to atmospheric. Each of the ducts has a constricted section into which is placed a pitot tube to measure

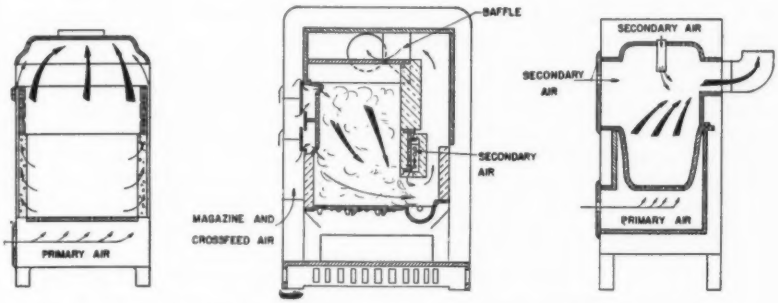

FIG. 2 *(left)*. MAGAZINE-TYPE HEATER WITH DEEP-FIRE-POT. FIG. 3 *(center)*. DIAGRAMMATIC VIEW OF THE CROSSFEED TYPE SMOKELESS HEATER WITH BAFFLE. FIG. 4 *(right)*. CONVENTIONAL SURFACE-FIRED HEATER WITH TWO ALTERNATE FORMS OF SECONDARY AIR ADMISSION

the rate of flow of air and a thermocouple grid to measure the temperature of the air. The floor is highly insulated and is not provided with thermocouples.

The principal unknown in regard to the measurement of any heat liberated within the room was the value of the overall heat conductivity of the materials forming the walls and ceiling of the room. This was determined by calibration tests for which an electric heater was used to insure that the heat output would be equal to the equivalent electric input. From these preliminary tests, the overall heat conductivity was obtained by subtracting from the heat input the measured heat in the circulated room air, and dividing the remainder by the measured temperature difference across walls and ceiling.

The room was provided with all the facilities for running burning tests of heaters from which heat losses in the flue gases and in the refuse could be directly determined. Knowledge of the heat input from the weight of coal burned and of the output as given by the room data thus made it possible to evaluate the unaccountable heat losses due to tar and soot in the flue gases.

DESCRIPTION OF HEATERS TESTED

Fig. 2 is a schematic diagram typical of *magazine-type*, deep-fire-pot heaters. These heaters consist of a cylindrical steel shell, lined with special refractory

shapes above the grate, and capped with a casting which usually also holds the firing door and the flue collar. Primary air passes through the grates, and gases may leave the fuel bed at various heights through side channels or ducts in the refractory or may leave at the top of the bed, the type of flow depending on the extent of caking in the fuel bed. Overfire air is admitted through slots in the firing door, or slightly below it, to promote secondary burning.

The two deep-fire-pot heaters used in the tests, and designated in this paper as Heater "A" and Heater "B", were selected on the basis of being both similar and yet different in the detail arrangement of the refractory linings; there was thus an opportunity to evaluate the influence of liner configuration on heater performance. There has been constantly the hope among designers of deep-fire-pot heaters that an improved design of liner would some day be developed that, through some novel change in the flow of air and gases in the heater, would insure dependable control of heater output and reasonably smokeless operation.

Fig. 3 is a sketch of the crossfeed type smokeless heater which utilizes alloy cast iron, steel, and refractory in its construction. Crossfeed or primary air enters the fuel bed above the grates; magazine air is admitted at the top of the fuel charge to prevent the formation of large coke pieces and thus assure self-feeding when caking coals are burned. The products of devolatilization from the magazine and from the combustion flow downward and across before passing under a secondary-air arch; at this point, secondary air is admitted through a continuous slot for the full width of the fuel bed to complete combustion. In Fig. 3, a baffle is shown in the flue-gas passage, over the magazine roof, to bring the hot flue gases in contact with the maximum area of top heat-transfer surface before their exit to the flue pipe.

Fig. 4 represents the conventional, surface-fired heater tested. Primary air is admitted through the grates and secondary air is admitted both through firing-door slots and through a *down-blast* pipe in the dome of the combustion unit. The heater is of cast-iron construction throughout.

METHOD OF TESTING HEATERS

Deep-Fire-Pot Heaters

Following preliminary trials on the test stand, outside the room, to determine the time and weight of each firing to reproduce fuel-bed conditions, each of the two deep-fire-pot heaters was successively tested in the calorimeter room. In these tests, each heater was subjected to a preliminary cycle and to four test cycles; the limitations observed in the successive tests were (1) a maximum applied draft of 0.060 in. of water and (2) maintenance of flue-gas temperatures of 1300, 1100, 900, and 700 F, respectively, during the four consecutive test cycles. The upper and lower flue-gas temperatures used had been established as the practical limits obtainable under controllable conditions with these heaters during the preliminary trials on the test stand.

The fuel capacity of Heaters "A" and "B", measured from the grates to within one inch of the top of the refractory liners, was 98 and 93 lb, respectively, of 3 × 2-in. coal. When 4 to 5 in. of residual bed thickness was allowed, the refueling capacities were 75 and 70 lb., respectively. Coal from the Island Creek seam was used for all tests of Heaters "A" and "B".

The duration of each test cycle was determined by the loss-in-weight method; the test was ended when the loss in weight was equal to the weight of fuel fired, minus the weight of ash in the fuel. Thus, tests of Heater "A" were ended when the weight loss reached 71.5 lb. Tests of Heater "B" were ended when the weight loss reached 66.5 lb.

During the period of devolatilization following the firing, the primary-air damper was closed to one-eighth open, the overfire-air damper was opened fully and the applied draft was adjusted to maintain the imposed limitation on flue-gas temperature. As the flue-gas temperature started to drop, the primary-air damper was gradually opened wider until further opening had no effect on flue-gas temperature. Then, the applied draft was gradually increased toward the imposed limitation of 0.060 in. of water as required to maintain the flue-gas temperature limit. The grates were shaken and the bed was poked to restore any further drop in temperature. The overfire-air damper was closed at the end of the devolatilization period, that is, when smoke emission ceased.

Readings for calorimeter-room data were taken every 20 min of (1) pitot-tube total pressures, (2) pitot-tube static pressures, and (3) air temperatures in the constricted ducts, and (4) of average voltage output of the differential thermocouples in the walls and ceiling. Readings were taken every half hour on weight loss and on applied stack draft. The barometric pressure and the wet and dry bulb temperatures were recorded every hour. Flue-gas samples were collected continuously over half-hour periods and then analyzed for CO_2, O_2, and CO. The smoke density, flue-gas temperature, and also the percentage of CO_2 in the flue gas were recorded on automatic instruments.

Smokeless Heater

Two series of tests were conducted on the smokeless heater in the calorimeter room. The first series was run with a baffle placed in the flue-gas passage. Following a preliminary cycle, the heater was subjected to three test cycles; the limitations observed were a maximum applied draft of 0.060 in. of water, and maintenance of flue-gas temperatures of 900, 775, and 650 F, respectively. The second series of tests was run immediately following, with the baffle removed, but under the same limitations as for the first series.

As in the tests of the deep-fire-pot heaters, the duration of each test cycle was determined by the loss-in-weight method, and 3 × 2-in. Island Creek coal was used as the test fuel. Since the refueling charge of the smokeless heater was set at 50 lb, each test cycle was ended at a weight loss of 47.5 lb.

The applied draft was varied between the maximum of 0.060 in. of water and a minimum of 0.020 in. of water to maintain the flue-gas-temperature limitation. Further reduction of the flue-gas temperature was obtained by closing the air-control dampers. The grates were shaken and the fuel bed was poked as often as was necessary to maintain the temperature prescribed. Readings taken were the same as those taken in the tests of the deep-fire-pot heaters described previously.

Conventional Surface-Fired Heater

The imposed limitations for each test of the surface-fired heater were that the flue-gas temperature would not exceed 900 F and that the applied stack draft would not exceed 0.060 in. of water.

Each of the tests consisted of at least three consecutive four-hour cycles. The alternate side-bank method of firing was used to minimize smoke emission. The heater was operated in such a manner as to prevent runaway operation after firing; the method used was that described previously for the deep-fire-pot heaters.

Both Illinois No. 6 and Pittsburgh coals in the $2 \times 1\frac{1}{2}$- and $2 \times 1\frac{1}{4}$-in. size, respectively, were used as test fuels. The readings of test data taken were the same as those in the tests described previously.

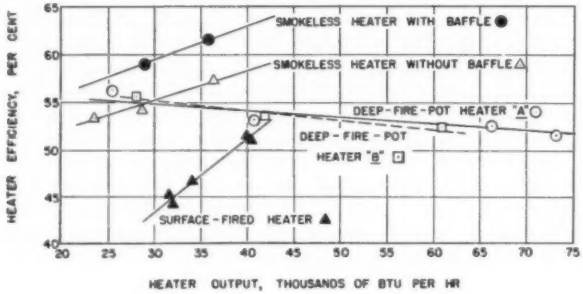

FIG. 5. RELATION BETWEEN HEATER OUTPUT AND HEATER EFFICIENCY FOR TWO DEEP-FIRE-POT HEATERS, A SURFACE-FIRED HEATER, AND THE CROSSFEED TYPE SMOKELESS HEATER, WHEN BURNING BITUMINOUS COAL

TEST RESULTS

Efficiency and Unaccounted-For Losses

Fig. 5 presents the results, in graphical form, of the efficiencies obtained during tests of the four heaters in the calorimeter room. Both the surface-fired heater and the smokeless heater showed an increase in efficiency as output increased. Contrasted to these results, the two deep-fire-pot heaters showed decreasing efficiencies with increased output. It is interesting to note that the efficiency curves for the two deep-fire-pot heaters are nearly identical over the full range of outputs tested.

With respect to the smokeless heater, Fig. 5 shows that a higher efficiency of the order of 5 percent was obtained with the baffle in place as compared to the tests without the baffle.

Fig. 6 shows the relation between heater output and average flue-gas temperature for the heaters tested. At the accepted limitation of 900 F on flue-gas temperature, outputs obtained for all heaters were within the range of 40,000 to 50,000 Btu per hour. However, the outputs of the deep-fire-pot heaters were considerably higher when the upper limit on flue-gas temperature was increased beyond 900 F.

Fig. 7 shows the unaccounted-for losses determined from the calorimeter-room data for all of the output tests run on the four heaters. The predominating

characteristic of the curves is the increase of the unaccounted-for loss to a maximum followed by a subsequent decrease as heater output increased. The unaccounted-for losses for the smokeless heater were consistently lower than for the surface-fired and deep-fire-pot heaters, and approached zero at the rating of the stove. In the range where outputs are equal, the unaccounted-for

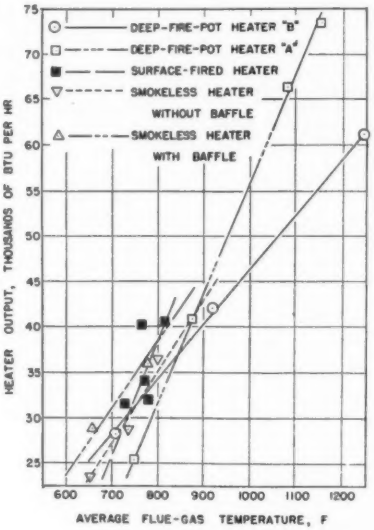

FIG. 6. RELATION BETWEEN HEATER OUTPUT AND AVERAGE FLUE-GAS TEMPERATURE OF TWO DEEP-FIRE-POT HEATERS, A SURFACE-FIRED HEATER, AND THE CROSSFEED TYPE SMOKELESS HEATER

losses for the surface-fired heater and for the two deep-fire-pot heaters are of the same general order.

The initial increase and subsequent decrease of the unaccounted-for loss, typical for all heaters, is considered to be brought about somewhat as follows: As the rate of burning is increased with accompanying increase in temperature of the fuel bed, the rate of devolatilization of the fresh charge increases, and the fraction of the tars that is not burned increases; but, finally, with further increase in rate, the temperature and the degree of turbulence increase to the point that an increasing fraction of the tars are ignited and burned with decrease in unaccounted-for losses.

Although the efficiencies of the two deep-fire-pot heaters are nearly equal at all outputs, the unaccounted-for losses were lower in the tests of Heater "B"

than for Heater "A" at the same output. The difference is believed to be the result of the increased burning of tars in Heater "B", because of the high available flue-gas temperatures at the same output. The higher flue-gas temperatures were probably the result, in turn, of the difference in refractory-liner configuration utilized by the two heaters, and, to a small extent, of the slightly smaller heat-transfer area of Heater "B". That both heaters had about the same efficiency for equal outputs resulted from the fact that the lower unaccounted-for loss of Heater "B" was offset by a correspondingly higher dry flue-gas loss.

Estimates made in the past of the unaccounted-for loss when burning high-volatile bituminous coal in space heaters have usually ranged from 15 to 20

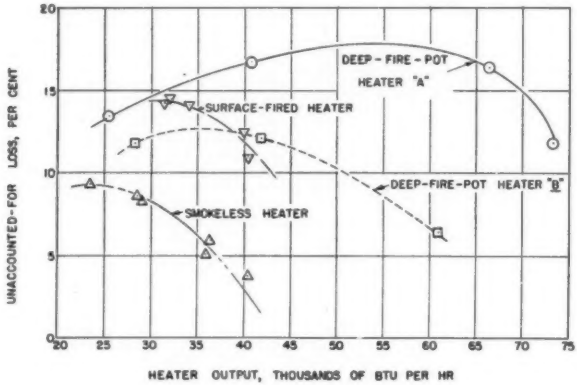

FIG. 7. UNACCOUNTED-FOR LOSSES DUE TO SOOT AND UNBURNED HYDROCARBONS IN THE FLUE GAS WHEN BURNING BITUMINOUS COAL IN SPACE HEATERS

percent of the total heat input. As shown by Fig. 7, these losses for overfeed-type heaters actually are somewhat less, even when at their maximum. In addition, the variation of the unaccounted-for loss at different burning rates for a given heater, and the extent of the differences between similar heaters, demonstrates clearly that the true-output rating of a heater can only be determined by the calorimeter-room method.

Comparison of Heaters

Table 1 presents a summary of the performance of each of the four heaters. These data were obtained from the performance curves previously discussed and compare the heat balances, burning rates, flue-gas temperatures, flue-gas compositions, and durations of test at an output of 40,000 Btu per hour for the four heaters. At this output, the flue-gas temperature, for all heaters, was below the 900 F temperature usually considered as a maximum not to be exceeded.

The table shows that the smokeless heater has an efficiency of 62.8 percent with a corresponding unaccounted-for loss of 3.0 percent; the efficiency of Heater "A" was 54.0 percent with a 16.6 percent unaccounted-for loss; the efficiency of Heater "B" was 54.0 percent with a 12.3 percent unaccounted-for loss; and the efficiency of the surface-fired heater was 51.0 percent with an 11.8

TABLE 1—COMPARISON OF HEATERS AT 40,000 BTU PER HOUR OUTPUT

HEATER	SMOKELESS WITH BAFFLE	DEEP-FIRE-POT HEATERS		SURFACE-FIRED
		"A"	"B"	
Coal	Island Creek	Island Creek	Island Creek	Pittsburgh
Size, in.	3 × 2	3 × 2	3 × 2	2 × 1¼
Calorific value, Btu per lb	14,180	14,190	14,190	13,970
Average burning rate, lb per hr	4.50	5.22	5.22	5.63
Duration, hr	11.4	14.1	13.3	4.0
Average flue-gas temperature, F	820	870	895	810
Average flue-gas composition, percent				
CO_2	8.6	9.8	9.1	7.4
O_2	11.3	9.0	9.8	9.8
CO	0.0	0.0	0.0	0.0
Smoke emission, Ringelmann No., minutes[a]				
Above No. 1	0	390	400	21
Above No. 2	0	250	176	18
Above No. 3	0	200	75	16
Above No. 4	0	65	20	12
Heat balance, percent				
Losses:				
Steam	4.6	4.7	4.7	4.7
Dry flue gas	27.6	24.1	27.3	29.7
CO	0.0	0.0	0.0	0.0
Ashpit	2.0	0.6	1.7	2.8
Unaccounted for	3.0	16.6	12.3	11.8
Heat utilized (efficiency)	62.8	54.0	54.0	51.0
Total	100.0	100.0	100.0	100.0

[a] Data taken from actual records of test cycle of each heater approaching most nearly the 40,000 Btu per hour nominal output.

percent unaccounted-for loss. It may be noted that the surface-fired heater and the deep-fire-pot heaters are about equal in regard to efficiency and unaccounted-for loss, respectively.

In addition to data on efficiency and unaccounted-for loss, the preliminary and room tests conducted during the course of this program provided an opportunity to compare all three types of heaters under similar operating conditions. The generally accepted merits and deficiencies of each type were well substantiated.

Control of the deep-fire-pot units in the room tests was difficult and was accomplished only by frequently adjusting the primary-air damper and the stack

draft. A positive pressure in the stack was needed, at times, to obtain low and medium rates of burning; since users will not tolerate gassing from the units under these conditions, the result is that a certain amount of overheating can be expected with these heaters. Attention to the fuel bed, either by shaking the grates or by poking the bed, was required three or four times during the latter portion of a cycle to maintain the desired rate of burning with caking coals. Table 1 shows that smoke emission was considerable.

The deep-fire-pot-type heaters do, however, have their advantages. Their large fuel capacity permits an appreciable lengthening of the interval between firings, as compared with that obtainable from the surface-fired heater. Also, the higher thermal head generated within the fuel bed promotes rapid pickup

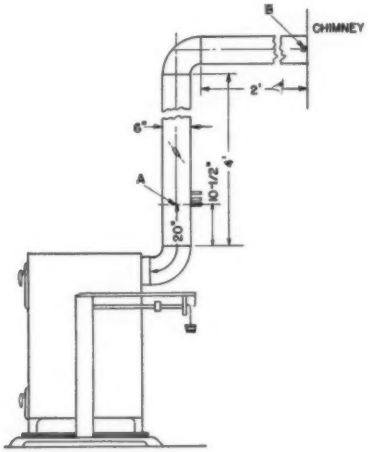

FIG. 8. TEST SETUP OF A TYPICAL COAL-FIRED HEATER INSTALLATION

and substantial heat output under what might normally be adverse chimney conditions.

The disadvantages of the surface-fired heater are essentially the same as those of the deep-fire-pot heaters, but on a smaller scale. Thus, there is only mild runaway operation, owing to the smaller weight of fuel fired, and thinner bed of fuel. Attention to the fuel bed is required only at the end of each cycle; the duration of each cycle at rating is four hours which compares unfavorably with the deep-pot units. Smoke emission on the same time basis as for the deep-fire pot heaters is of the same order of intensity for smoke densities of No. 3 or higher.

The tests of the smokeless heater showed longer periods between firings than are obtained with the surface-fired heater, but these periods are slightly shorter than those obtained with the deep-fire-pot heaters. There is no runaway operation with the smokeless heater, because of the use of a stack-temperature limit

control. Output is relatively constant and control of the output, whenever required, is easily obtained by adjustment of the stack draft and slight adjustment of the primary-air damper. Attention given the fuel bed, either by shaking the grates or by poking the bed, was required only once during each test cycle. In addition, efficiency is high and there is negligible smoke emission.

The main disadvantage in the operation of the smokeless heater is its relatively slow pickup, after firing on a cool residual bed. The duration of time required to reach rating varies; the higher the rate at which the residual fuel bed is burning, the quicker rating is reached. This disadvantage is compensated for, under ordinary heating conditions, since the stable output and long attention periods of the smokeless heater will keep the space being heated at a comfortable temperature without the necessity of quick pickup.

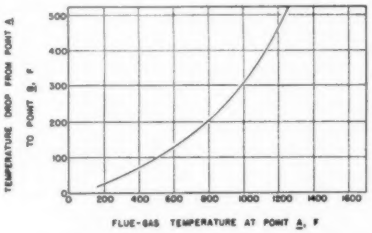

FIG. 9. RELATION OF TEMPERATURE DROP FROM POINT A TO POINT B TO FLUE-GAS TEMPERATURE AT POINT A FOR A TYPICAL HEATER INSTALLATION

HEAT GAIN FROM FLUE PIPE

The efficiencies shown in Table 1 are based on a flue-gas temperature which is measured 20 in. from the flue collar of the heater along the centerline of the stack. However, in most heater installations, useful heat is also supplied from the remainder of the flue pipe connecting the heater to the chimney, and, to some extent, from the chimney itself. This factor is taken into account in designing coal heaters. The furnace pipe not only vents the gases, but, for over 100 years, has been the lowest in cost, simplest, and most effective heat exchanger that has been conceived for the purpose. Because of a higher temperature, the flue pipe on a coal heater is a more important heating surface than flues for other fuels, notably gas.

A series of tests was, therefore, run with the aim of determining the amount of useful heat supplied from a typical arrangement of flue pipe.

Test Setup and Procedure

Fig. 8 shows the arrangement used for experimental work on a flue-pipe setup which is typical of many coal-fired-heater installations. To measure the heat given off by the pipe, aspirating-type thermocouples were placed at Point A, 20 in. above the flue collar, and at Point B, the entrance to the chimney, to measure the temperature drop of the flue-gases from the smokeless heater. In

this setup a 6-in. diameter, black stove pipe was used. Tests were run at high, medium, and low rates of burning.

Test Results

Fig. 9 is a plot showing the results of the tests to determine the relation between the drop in temperature from Point A, of Fig. 8, to the entrance to the chimney and the flue-gas temperature at Point A.

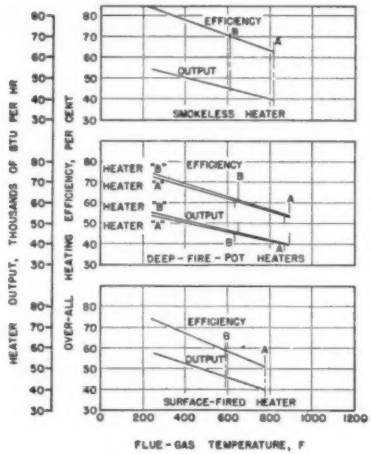

FIG. 10. RELATION OF OUTPUT AND OVERALL HEATING EFFICIENCY TO FLUE-GAS TEMPERATURE FOR THE TYPICAL INSTALLATION OF COAL-FIRED HEATERS OPERATING AT A NOMINAL OUTPUT OF 40,000 BTU PER HOUR

These data are, in the strict sense, applicable only to the particular type of heater used when operating under similar test conditions. However, the data are believed to be very nearly those that would be obtained with other types of equipment operating at the same burning rates.

OVERALL HOUSE-HEATING EFFICIENCY

An overall house-heating efficiency, higher than the nominal efficiency, can be calculated for heaters when the obvious contribution from heat lost by radiation from the flue pipe is taken into account, and also when it can be assumed that heat from the chimney will find its way into the dwelling.

Fig. 10, which can be used for the determination of the overall house-heating efficiency, is a plot showing the relation of output and overall heating efficiency

to flue-gas temperature for the typical installation of a coal-fired heater shown in Fig. 8. Points A and B on the graph are the corresponding flue-gas temperatures at Points A and B in Fig. 8.

Fig. 10 shows that, when the smokeless heater is operating at a nominal output of 40,000 Btu per hour, with an efficiency of 62.8 percent, the flue-gas temperature at Point A is 820 F. The temperature at Point B, where the gases enter the chimney is, however, 610 F, a drop of 210 deg. Thus, the room has gained an additional amount of heat which corresponds to this drop in temperature, an amount equal to 5000 Btu per hour. This represents an increase in output of useful heat from the heater to 45,000 Btu per hour and an increase in efficiency to 70.4 percent. Similarly, the output of Heater "A" would be increased to 45,200 Btu per hour with an efficiency of 61.0 percent; the output of Heater "B" would be increased to 45,600 Btu per hour with an efficiency of 61.8 percent; and the output of the surface-fired heater would be increased to 46,300 Btu per hour with an efficiency of 59.2 percent.

If all the heat given up by the flue gases, as they pass on up the chimney, is considered useful, the total heat utilized and corresponding overall house-heating efficiency can be obtained from Fig. 10, by determining the flue-gas temperature at the house-roof line, and reading directly from the curves the output and efficiency corresponding to this temperature value.

Since the data given in Fig. 10 are based on heater outputs at or near the maximum rating of the heaters, efficiencies for overall house-heating taken from the figure cannot be construed as being representative of the seasonal efficiency that would be obtained; also, intangible factors such as operator skill are not taken into account. The data are applicable for heating-system design purposes, however, because heating-load calculations are based upon the extreme temperature conditions expected, and the heaters would be operating at high-output rates under such conditions.

ACKNOWLEDGMENT

Acknowledgment is made of the contributions and suggestions of Elmer R. Kaiser, assistant director of research, and James R. Garvey, development engineer, Bituminous Coal Research, Inc.; of the work of Paul Vyff, former research engineer at *Battelle Memorial Institute,* in the development of the calorimeter room; and of the contributions of Charles W. Porterfield, research engineer at *Battelle Memorial Institute,* to the work on heat gain from a flue pipe.

Acknowledgment is also made to H. J. Rose, vice president and director of research, Bituminous Coal Research, Inc., and to the Hand-Fired Heating Equipment Committee of the Technical Advisory Board of Bituminous Coal Research, Inc., for their interest and support of the program.

DISCUSSION

C. L. SAPP, Wilmington, Ohio (WRITTEN): In discussing Fig. 5, the authors make an interesting comment in connection with the contrasting efficiency results between the two deep-fire-pot heaters, and the smokeless and surface-fired heaters.

What analysis would account for the decreasing efficiencies of the two deep-fire-pot heaters with increased output, when the smokeless heater showed some increase, and

the surface-fired heater a rather sharp increase in efficiency? Would heating-surface grate-area ratio, method of introducing secondary air, or variations in flue outlets size contribute to this contrast?

The method of determining the increase in output from the surface-fired heater is not quite clear. If the heat gain from the flue pipe is 5000 Btu in the case of the smokeless heater, with a temperature at Point A of 820 F, why is the heat gain of the surface-fired heater 6,300 Btu with a temperature at Point A of 810 F?

In a paper given by R. C. Cross at the A.S.H.V.E. Annual Meeting in 1943[1] efficiencies of the surface-fired space heater tested were of the order of 55 to 56 percent with a draft limitation of 0.02 in. Do the authors of this paper feel that an increase of draft limitation to 0.06 in. would be responsible for a reduction of 10 percent in heater efficiency? Or would the use of the calorimeter room provide that much more accuracy in measuring heat output?

It would also be interesting and valuable information to learn whether field tests show that space heater users obtain the relative outputs and efficiencies indicated by the tests. Would the slightly more complicated control mechanism, and possibly more careful firing attention required for the smokeless and deep-fire heater tend to discount the higher efficiencies obtained under laboratory conditions?

OTTO J. RESS, Milwaukee, Wis. (WRITTEN): The methods of determination of the losses due to smoke and unburned components of the fuel are of considerable interest. In the past many references have been made to the fact that the actual available heat output of heating appliances is not only the bonnet output of the appliance but also the sum of such factors as heat regained from the smoke pipe and chimney. References have also been made to an assumed percentage of unaccounted for loss in the nature of unburned components of the fuel. The magnitude of these regains and losses could not be determined through use of the conventional indirect method of flue loss determination. The presence of these various factors has been recognized but their determination as to magnitude has not been available and particularly not in regard to tests with different appliances and operating at different rates.

In the method under discussion, the unaccounted for losses, which are assumed to to be the measure of the unburned fuel, are determined by the difference between the output of the unit as measured by the flue loss method and the output obtained through the direct method as measured by the calorimeter room. Accurate control must be kept to prevent the entrance of heat from the smoke pipe into this determination. The question arises as to the location of the stack temperature measuring point. If the stack temperature in the flue loss determination were taken at the flue collar of the appliance, the heat dissipated from the smoke pipe into the enclosure would result in a smaller difference between the flue loss of the unit and the calorimeter room reading with a corresponding reduction in the calculated unburned fuel loss. However, if the stack temperature determination was taken at the ceiling of the calorimeter room the effect of the heat dissipated from the smoke pipe into the enclosure would be accounted for in the flue loss and leave a difference between that figure and the room calorimeter reading as chargeable entirely to unburned fuel.

The possible effect of chilling of the smoke pipe due to the action of a check damper or a barometric draft regulator would have a tendency to reduce the heat supply to the room from the smoke pipe. This situation, of course, would not be true at times of maximum heater output, unless a barometric draft control was used. It would also be possible with the arrangement as described in this paper to conduct some very interesting tests on the effect of the heat which is removed from the enclosure due to the action of the barometric draft regulator. A series of tests could be conducted in which the barometric draft control is placed exterior to the enclosure

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[1] Performance Characteristics of a Coal-Fired Space Heater, by R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 99).

and a similar series of tests conducted in which the maintenance of a constant draft is effected through the use of a barometric draft regulator pulling makeup air and room heat directly from the enclosure.

The point of heat gain from the flue pipe is very well presented. The flue pipe does present a definite low cost and simple type of heat exchanger. However, it is interesting to note that in the 6 ft of 6-in. smoke pipe used in this test credited with a heat regain of 5,000 Btu, that the total surface represented by this smoke pipe is equal to 9.42 sq ft of heating surface. On the basis of the 5,000 Btu heat regain from this surface, the heat transmission per sq ft of smoke pipe is 530 Btu per sq ft. This figure of 530 Btu per sq ft of heating surface is definitely low in comparison with figures used for calculating the transmission of heating surfaces of the direct fired heaters themselves. One possible source of this, no doubt, is that the smoke pipe rising as it does vertically above the heater and then along the ceiling is in a zone of high temperature air in which the maximum effectiveness of this heating surface is not utilized.

The material contained in this paper in giving a definite value to factors which in the past have been placed at arbitrary percentages of the total output of the heating appliance is of definite interest and presents opportunities for clearer understanding of the sources of heat available for heating of the structure with certain types of appliances.

E. A. ERICKSON, Chicago, Ill. (WRITTEN): The authors are to be commended for presenting an excellent paper which is of practical value to fuel engineers and testing laboratories. For years, these engineers and laboratories have been using values obtained from their own experience in evaluating the unaccounted stack losses of domestic heating appliances when burning the various types of solid fuels.

That the efficiency curves for the two deep-fire-pot heaters, as shown in Fig. 5, were nearly identical over the full range of outputs was very interesting. These results tend to show the fallacy of claims for improved performance because of some ingenious liner design. It was even more surprising to note that the characteristics of efficiency for the surface-fired heater differed from those of the two deep-pot heaters. It would have been expected that the efficiency characteristics for both these types of heaters would be similar; that is: increasing with increased output up to the designed output of the heater.

Fig. 6 shows that the maximum output of 40,000 Btu per hour within test code limitations appears independent of heater design. If this is true, then it also appears that some formula, similar to that developed by the University of Illinois for rating warm-air furnaces, is in order, using the area of heat transfer surface as a basis. In line with this, it would be interesting to know the burning rates per sq ft of grate area for the conventional heaters tested.

In connection with Fig. 7, the authors present several reasons for the differences in unaccounted-for losses for Heaters "A" and "B". I should like to ask if shielded thermocouples were used to measure the temperatures of the flue-gases in the tests on the conventional heaters. In my experience in this work, I have noticed differences of up to 200 F between use of shielded and unshielded thermocouples in the high temperature ranges given in the paper. The fact that Heater "B" showed higher flue-gas temperatures with correspondingly lower unaccounted-for losses appears to warrant this question. A temperature difference of only 80 F would decrease the dry flue gas loss about 2.7 percent and correspondingly increase the unaccounted-for loss a like amount. If such were the case in these tests, the unaccounted-for losses for the two deep-pot heaters would be more nearly of the same order.

The authors state that the users of deep-pot heaters can expect a certain amount of overheating because of lack of control on burning rate with the deep fuel bed. I have noted in my observations on several surveys that the users do not fill these heaters to capacity. Apparently, they have learned a method of control to prevent

overheating. It would be interesting to see some results with respect to efficiency, unaccounted-for loss, and smoke performance with the heaters operated at only half-fuel capacity.

In conclusion, I would like to comment on the gain in overall house-heating efficiency with respect to the flue pipe and chimney. No remarks were given in the paper regarding the use of a barometric damper which is recommended for nearly all heater installations for control of burning rate and output. Because of the dilution of hot flue gases with the warm room air, the heat gains mentioned in the paper could be nullified. Results of tests have shown heat losses from 2000 to 7000 Btu per hour, depending upon the magnitude of the draft, as escaping through the barometric opening to the stack.

P. R. ACHENBACH, Washington, D. C.: I am very much interested in this paper because it describes tests similar to those made at the National Bureau of Standards during the last war on a large number of space heaters, some burning anthracite and some burning bituminous coal. We also used a calorimeter room for this purpose which was, I believe, a predecessor of that now used at *Battelle Memorial Institute.*

There are several questions that come to my mind about the test procedure. Fig. 1 in the paper indicates that a bellows-type joint was used in the smokepipe to permit free movement of the scale platform during the test in conjunction with a vertical smokepipe connection whereas the data in Fig. 9 on flue-pipe heat losses were obtained with an S-shaped or Z-shaped smokepipe. Are the data obtained from the Z-shaped connection applicable to the straight smokepipe and bellows arrangement used for the calorimeter tests?

Are the heater outputs plotted in Fig. 5 values obtained from the calorimeter measurements and have these values already been corrected for the heat loss of the smokepipe from the station of flue-gas temperature measurement 20 in. from the flue collar to the ceiling of the calorimeter room? That is, do the unaccounted-for losses plotted in Fig. 7 represent a difference between the direct and indirect efficiencies of the heater both related to the station of flue-gas measurement 20 in. from the flue collar?

I would like to inquire whether or not complete flue-gas analyses were made at any station in the smokepipe. It is possible, of course, by measuring the total unburned hydrogen and carbon atoms in the flue gases to evaluate all of the incomplete combustion loss except that represented by soot and tar that might be deposited in the sampling line or its filter. Table 1 shows that there was no carbon monoxide in the flue gases for any of the four tests reported. Our tests of magazine type coal heaters at the National Bureau of Standards showed that there were, with almost no exceptions, measurable percentages of carbon monoxide, hydrogen, and unburned hydrocarbons in the flue gases with both anthracite and bituminous coal causing incomplete combustion losses ranging from 4 to 18 percent of the heating value of the fuel. When these incomplete combustion losses were taken into account, the direct and indirect efficiencies agreed within three percent or less in all tests but two when the disagreement amounted to 5 percent. That is, our experience indicates that a fairly good agreement between indirect and direct efficiencies and between indirect and direct outputs can be obtained when a complete flue gas analysis is made with a precision flue-gas analyzer. It further indicated that the flue losses that could be attributed to soot and heavy tarry materials was of a magnitude less than the precision of the other measurements.

R. K. THULMAN, Washington, D. C.: I would like to ask the author if the two deep-fire-pot heaters were jacketed. I believe the smokeless heater is a jacketed type of heater.

I would also like to ask the author; does the use of a jacket account for any difference in efficiency between the smokeless heater and the deep-fire-pot heaters?

By way of comment I would like to suggest that the calorimeter method of measuring the output of heaters of this type might be questioned in view of the practical effects that we get in the use of space heaters which produce a very high temperature gradient within the room; I dare say that the temperature measurements within the calorimeter room might have been fairly uniform because of the forced distribution of air within it.

AUTHORS' CLOSURE: The comments and questions presented by the discussors are indicative of the widespread interest in technical problems associated with the rating of space heaters. Despite the extensive use of coal-burning heaters, few published data on their performance are available in the literature, and much in the way of systematic testing is still required. Examples of such additional tests would be a series designed to evaluate the effect of barometric dampers as suggested by Mr. Ress and by Mr. Erickson, and field tests under controlled conditions as suggested by Mr. Sapp.

Mr. Achenbach's statement regarding the close correlation between direct and indirect efficiencies when a complete gas analysis was made during work at the National Bureau of Standards is a valuable contribution; attempts to make a similar comparison have not been made at *Battelle Memorial Institute*. When the calorimeter room was constructed at *Battelle Memorial Institute* for Bituminous Coal Research, Inc., it was decided that the room method was the only positive means of establishing beyond doubt the magnitude of total unaccounted-for losses associated with the burning of bituminous coal. Tests with anthracite, for which unaccounted-for losses are negligible, showed that heater output as determined by the room data checked within 2 percent of the heater output as determined indirectly.

In answer to Mr. Achenbach's and Mr. Ress' questions concerning flue-gas temperature measurement, a thermocouple was placed in the stack 20 in. above the heater flue collar, as is recommended in most heater test codes. All calorimeter room results of output, efficiency, and unaccounted-for loss were based on this temperature. In instrumenting the calorimeter room, a second flue-gas-temperature thermocouple was placed at the point in the stack where the flue gases pass at the level of the ceiling. Thus, it is possible to calculate the heat loss to the room between these two levels. This amount of heat is then subtracted from the total heat measured by the room to give the heater output, including the first 20 in. of flue pipe. Data on heat gain from the typical flue pipe installation described in the paper are based on the temperature drop in that portion of the pipe which is beyond the first 20 in. and may be applied to any standard type of flue pipe configuration.

Replying to Mr. Erickson's question, only the bare type of thermocouple was used. Since the thermocouples were exposed to flame during portions of the test cycles of the deep-fire-pot heaters, it is possible that this would affect the determination of dry flue-gas loss as suggested. However, the authors doubt that the difference between the two types of deep-fire-pot heaters can be explained in this manner, since, from observation, the flame travel patterns in each appeared to be similar.

Regarding Mr. Thulman's questions on the jacketing of the heaters, all heaters except the conventional surface-fired unit were of the radiant type. Tests of a smokeless heater in the calorimeter room, some years ago, both with and without a jacket, showed no significant difference in heater output or efficiency.

Mr. Sapp has asked several questions regarding the interpretation of the results presented in the paper and their application to practical field conditions. The deep-fire-pot heaters have only a limited area for heat transfer that is not protected by refractory; this area consists of the top and flue pipe. As pointed out by Mr. Ress, the flue pipe is not so effective as direct-fired heater surfaces in transferring heat, and the decreasing efficiency of the deep-fire-pot heaters, which results from increasing dry flue-gas losses, can probably be explained at least in part by this fact. The difference in heat gain from the flue pipe for the various heaters comes about because the mass flow of gases varies for each as well as the temperature drop in the flue pipe.

The lower efficiencies for a surface-fired heater presented in this paper, as compared to those given by R. C. Cross, are a result of higher dry flue-gas losses obtained for the heater tested in the calorimeter room. They are due in turn to the higher flue-gas temperatures and the lower CO_2 obtained at *Battelle Memorial Institute*. Since Mr. Cross was on the side of conservatism in assuming a 20 percent unaccounted-for loss, the true efficiency of the heater described in his paper is probably even higher than claimed. The performance characteristics of the two heaters are probably different because of differences in heat-transfer area or arrangement; moreover, there were probably differences in the test procedure. For example, the secondary-air damper was wide open during the Battelle tests, but the same area for admission of secondary air may not have been used by Mr. Cross.

1411

No. 1411

THE MEASUREMENT OF SMOKE

By K. O. Beatty, Jr.* and J. E. Deas, Jr.,† Raleigh, N. C.

SMOKE emitted from a combustion process may indicate incomplete combustion, a direct economic consideration, and may also produce a community nuisance. This latter aspect has been the subject of a number of papers [1,2]. These problems have stimulated investigators to develop a number of methods for evaluating the extent and nature of smoke. Some of these workers have been concerned with research on combustion processes and others have been interested in control and abatement of atmospheric pollution. All however, have sought some adequate procedure for assigning quantitative values to smoke.

Quantitative measurement of smoke presents a number of virtually insoluble problems. This is apparent from the different characteristics which may make smoke undesirable. Poorly operated furnaces may produce sufficient smoke to create an undesirable atmospheric haze. Deposits of soot in the flues and stack may be of more direct significance to the plant owner. Furthermore, neither the amount of smoke visible nor the soot deposited are measures of the efficiency of combustion of the fuel. A finely dispersed smoke causing appreciable haze may leave no visible deposits anywhere. On the other hand, combustion conditions which rapidly clog the flue and stack with soot may cause only a faint gray stream at the chimney top.

An experimental procedure which measures accurately any one of these characteristics of smoke cannot, of necessity, be expected to give a direct correlation with a procedure which measures some other equally important aspect. There is not, nor can there be, any single solution to the problem of quantitative measurement of smoke. Concrete evidence of the ambiguities resulting from the use of different methods is offered later in this paper.

A brief discussion of some of the various methods now in use, their principles and what they actually measure, seems desirable and is given in the next section. Modifications of three of these methods have been used simultaneously and the

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[1] Exponent numerals refer to References.

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data so obtained allow direct comparisons of results of different instruments on the same smoke, thus permitting some interpretation of their relative significance. These results are given in the latter part of this paper.

PRINCIPAL METHODS OF MEASURING SMOKE

Smoke indicators may be divided into three groups according to the *A.S.M.E.* Power Test Code of 1945 [3] as follows:

1. Those which are intended for measuring smoke density, according to some scale, either within the stack itself or as it issues from the stack.

2. Those which give a quantitative measure of the degree of smoke pollution in an atmosphere.

3. Those which are primarily operating aids and do not measure smoke density except indirectly.

It is only the first of these groups that is to be considered here, although the methods of the other two groups are basically the same. The methods in common use may be classed as either qualitative or quantitative procedures. The classification is not distinct since without a clear definition of what smoke is, no method can be said to be truly quantitative. However, those procedures which attempt to make measurements under conditions which can be reproduced at the will of the operator, have been called quantitative.

QUALITATIVE METHODS

Ringelmann Charts. It is a common but incorrect belief that the amount of soot and flyash in smoke is proportional to its visual density. This is approximately true only when atmospheric conditions, chimney diameter and fuel rate remain constant. Notwithstanding these limitations, the visual density of a smoke has become accepted widely as a legal criterion for maximum allowable smoke.

Probably the first widely used standard of visually measuring smoke was the familiar Ringelmann chart method. The Ringelmann charts are a series of cards, ruled as shown in Fig. 1, so that at a distance they appear as shades of gray. The density of these charts covers the range of solid white to solid black through four intermediate shades of gray. These charts are visually compared with the effluent smoke to give values from 0 for no smoke to 5 for dense black smoke. The chart (in the form of a card having 200 or 300 squares of the type shown) is set in a direct line between the chimney and the observer at a distance of about 50 ft from the observer. The observer then makes visual comparison of the smoke directly across the top of the stack with the apparent shades of the Ringelmann cards as observed at a distance of 50 ft.

This method has gained widespread use since it involves inexpensive equipment and evaluates apparent density of the effluent smoke, which is the feature most likely to be criticized by the public. It is desired in most cases to keep the apparent density of the smoke entering the air to a minimum and therefore, city smoke ordinances have generally specified a certain Ringelmann number as the maximum density allowable.

There are certain disadvantages, however, in this visual smoke density evaluation [4].

1. The apparent density of the smoke depends on the amount of light coming through the smoke to the observer, and hence, it depends on the nature of the background.

2. It is influenced by the wind velocity. When the wind is blowing strongly, the smoke is scattered widely and appears very thin; however, when the air is still, the smoke accumulates, giving a more dense appearance.

3. High relative humidity tends to intensify the apparent density of any smoke dispersion.

4. The size of the chimney from which the smoke issues causes a marked change in the apparent density of a smoke. The larger the chimney, the denser the appearance of the smoke if other factors are constant.

5. The smoke inspector has a problem of judging the comparative density of two widely spaced surfaces, *i.e.*, the charts and the smoke.

With compensations for these objections, the Ringelmann method may be satisfactory for municipal ordinance control, but it is of little value for purposes of quantitative comparison.

Umbrascope. The umbrascope was developed to overcome some of the more obvious limitations of the Ringelmann charts. In its use a small sighting tube is arranged so that half of the field of vision may be covered by semicircular plates of clouded glass. The degree of darkness can be regulated by inserting up to four glasses in series. A direct comparison can then be made between the clouded glasses and the smoke as viewed through the open half of the field of vision.

Details of both of these methods together with variations of them are given in the *A.S.M.E.* Power Test Code [3].

QUANTITATIVE METHODS

Filter Spot. Particles of smoke dispersed in a gas can be collected by means of an appropriate filter medium. If a fixed volume of flue gas is drawn through a filter, such as white filter paper, the density of the spot produced by the deposited smoke particles can be used as a measure of smoke density in the stack. The spot thus obtained is usually compared with various standard spots ranging from white for no smoke to black for dense smoke. In one system, a number is assigned to each standard spot, ranging from 0 to 9 by units of ½ (19 standard spots total). For combustion processes which produce black smoke spots such as those burning coal or petroleum products, this method of smoke measurement has been used with considerable success[5]. However, the apparent intensity of the spot is a function of color as well as quantity of smoke deposited. White smoke particles would fail to show up on the filter, and intermediate color shades would show up to a lesser degree than would the same quantity of black smoke particles. Particle size also affects the results obtained with this method of smoke measurement. Very fine particles may be only partially collected on the filter and the shade of the spot obtained may be different from that for an equal weight of larger particles.

In recent years several smoke meters have been designed to make use of this principle of collecting smoke particles on a filter. The North Carolina State College, Department of Engineering Research, has fabricated an apparatus designated the *N.C.S.* smoke meter which employs this principle. It is discussed in detail in the section *N.C.S. Smoke Method.*

Although visual comparison between the spot and a special scale is common, alternative procedures such as photoelectric devices and weighing have been used.

Deposit Methods. It is a well established fact that smoke particles tend to deposit on surfaces in contact with the smoke, particularly if these surfaces are

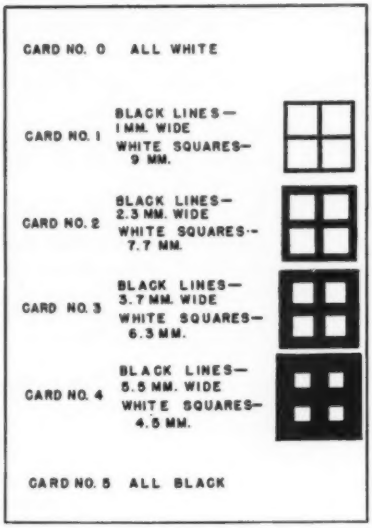

CARD NO. 0 ALL WHITE
CARD NO. 1
BLACK LINES— 1 MM. WIDE
WHITE SQUARES— 9 MM.
CARD NO. 2
BLACK LINES— 2.3 MM. WIDE
WHITE SQUARES— 7.7 MM.
CARD NO. 3
BLACK LINES— 3.7 MM. WIDE
WHITE SQUARES— 6.3 MM.
CARD NO. 4
BLACK LINES— 5.5 MM. WIDE
WHITE SQUARES— 4.5 MM.
CARD NO. 5 ALL BLACK

FIG. 1. SPECIFICATIONS FOR RULING OF RINGELMANN CHARTS

cooler than the smoke. The rate of deposition on a surface can be used as a measure of smoke density in a flue or stack under certain conditions.

One method for determining the rate of deposition is to allow smoke to settle on a clear glass rod for a given length of time. The density of the deposited smoke particles is then measured by means of a photoelectric meter calibrated to indicate the percent of light transmitted, based on 100 percent transmission for a clean glass rod. The *Institute of Cooking and Heating Appliance Manufacturers* has developed such a meter, and it is described in a later section under the heading *I.C.H.A.M. Method.*

Another method which might be used for determining the rate of deposition of smoke particles on a surface suspended in a smoke stream is a sensitive weighing device. A Jolly balance may be used to weigh a screen or other suitable surface suspended in the gas stream through a small hole in the stack. The rate of increase in weight of the suspended surface due to smoke deposition will be a function of the density of the smoke. This method has not, to the

knowledge of the writers, been used for measuring smoke density although the principle has been used in testing anti-sooting agents[9].

Light (Radiant Energy) Transmission Methods. A characteristic of smoke which lends itself readily to quantitative measurement is its obscuring effect on light. When a beam is directed through a gaseous dispersion of smoke particles, some of the light is absorbed by the particles, some is scattered at various angles, and some passes through undeflected. If a means is provided for measuring the light transmitted through the smoke, the decrease in light intensity from that obtained without smoke is a measure of the smoke density.

The light transmitted through a dispersion such as smoke is a complex function of particle size, concentration and nature[6]. A very minute amount of material appropriately dispersed may produce a highly opaque smoke whereas a relatively large amount suspended as coarse particles may give high transmission factors. The method is thus a measure of the ability of the smoke to obscure light and not necessarily a measure of the density of the smoke as it would appear to an observer. To an observer, the appearance of smoke is a function of color and background. Smoke viewed by a combination of reflected and transmitted light may appear to have a different density than the same smoke viewed entirely by transmitted light.

One device for measuring light intensity is a photoelectric cell. In practice, a constant intensity light source is mounted so that a light beam, directed through a section of the smoke pipe, strikes the photoelectric cell. Usually for a large industrial stack, the beam is directed transverse to the flow of gas. In smaller stacks the beam may be directed longitudinally through an offset section of the pipe through which the entire products of combustion must pass. The light received by the photoelectric cell is measured by the output of the cell as read on a microammeter calibrated directly in microamperes or in arbitrary smoke units.

A *photoelectric* smoke meter similar to one developed by Underwriters' Laboratories, Inc. with certain modifications has been successfully employed in the laboratories of the Department of Engineering Research for making smoke measurements. This instrument is described more fully in a later section under the heading *Photoelectric Smoke Meter.*

Another transmission method of measuring smoke density is based on the bolometric principle of measuring minute quantities of radiant energy. A source of radiant energy, usually an incandescent light, is mounted at one side of the smoke stream and the radiant energy transmitted is collected at the other side. The bolometer differs from the photoelectric cell in that it measures total radiation rather than light intensity alone. Since the majority of energy particles emitted by an incandescent light is in the infra-red region, the amount of absorption observed may be different from that observed with a photoelectric cell. A smoke density recorder, making use of the bolometric principle, has recently been placed on the market.

The advantages of the radiant energy transmission methods of measuring smoke density may be summarized as follows:

1. They give a continuous check on smoke density and are easily adapted to recording instruments.

2. When adjusted and maintained properly, they give a high accuracy and sensitivity to changes in smoking rates.

3. They can be equipped to alert an operator when the maximum allowable smoke density has been reached, so that steps can quickly be taken to correct the condition.

A disadvantage common to both the photoelectric and bolometric methods is the tendency of soot and dust to deposit on the light source and the collecting cell windows. The *photoelectric* method has the further disadvantage that the output of the cell tends to decrease upon prolonged exposure to light.

Electrical Precipitation. When smoke particles, fumes or liquid droplets, entrained in a gas, are given a high negative electrical charge, they will adhere to a positively charged plate. By using a suitable means of collecting these adhered particles, the entrained solid and liquid particles can be effectively removed from effluent gas streams.

The electrical precipitator makes use of this principle for a wide range of dust removing purposes. The precipitator in its simplest form consists of a wire

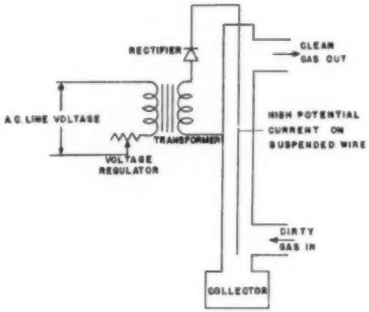


FIG. 2. DIAGRAM OF ELECTRICAL PRECIPITATOR

suspended in a pipe through which the gas flows. A high tension direct current passes from the wire through the gas (negatively charging the particles present) to the pipe which serves as the positively charged collector plate. The particles rapidly move toward the positively charged surface and adhere; however, if too many of these adhered particles accumulate, some of this material may flake off and become again entrained in the gas and thus be carried to the atmosphere. Therefore, the usual practice is to provide for intermittent removal of the deposit from the positive plate.

The primary source of current generally used is 110 to 550 volts a-c power stepped up by means of a transformer to a range of 15,000 to 25,000 volts[7] for small scale laboratory work and 50,000 to 75,000 volts [8] for industrial size units. The current is then rectified to unidirectional, high potential current, suitable for precipitation. A diagram of the essential elements of an electrical precipitator is given in Fig. 2. In laboratory devices, aluminum foil may be used to line the pipe so that the charged particles collect on the aluminum foil from which they can later be removed.

Portable electrostatic samplers are available and in wide use for smoke and dust contamination work of all sorts. One of their principal advantages is the

high percentage of recovery of even extremely minute particles. The material recovered is in a form well suited for further examination as to its physical and chemical nature. To the knowledge of the authors, this principle has not been employed in the routine measurement of smoke resulting from the combustion of fuels.

Counting of Particles. Smoke particles generally range in size from 0.2 to 1.5 microns (1 micron = 1/1000 millimeter). A knowledge of the number of particles in a given volume of gas together with the average particle diameter constitutes a good measure of the density of a smoke. The most widely used research tool for making smoke counts is the ultramicroscope because the particles are frequently below the range of the ordinary microscope. The apparatus consists of a suitably lighted sampling chamber and an ultramicroscope fitted with a slit or eyepiece aperture which restricts the vision to a known area. From the depth of focus, the area of the aperture used and the particle count, the number of smoke particles per unit volume can be calculated. For operation, a sample of smoke is drawn into the sampling cell and a number of successive counts are made. As many as fifty counts may be necessary in order to have a reliable coverage.

A method for collecting the smoke particles so that a count may be taken at leisure has been proposed by R. Whytlaw-Gray, W. Cawood and H. S. Patterson [10]. By this method a sample of the smoke is trapped in a small space of known volume between two glass plates in a special holder. The smoke particles are allowed to settle out on the lower glass plate which is then removed and placed under an ultramicroscope where the particles may be counted.

Particle size measurements present an appreciable problem because their range is frequently below that permitting direct or photographic microscopic examination. In many studies, the particle diameters are merely estimated and particle count is used as the basis of comparison between smokes.

METHODS USED AT NORTH CAROLINA STATE COLLEGE

Since it was felt that no one method of measuring smoke would give complete information, three methods were chosen for investigation, each being based on a different principle. They are described in some detail in the following pages and a comparison is given of results obtained with all three methods under a variety of combustion conditions.

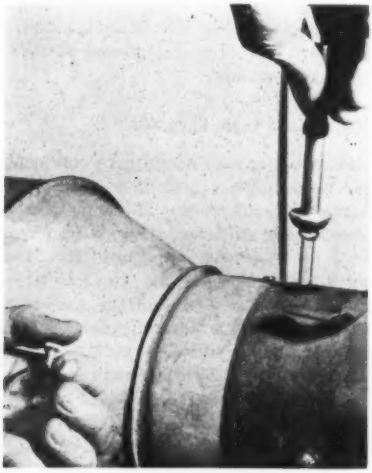
I.C.H.A.M. Method. This method receives its name from the initials of its sponsor, the *Institute of Cooking and Heating Appliance Manufacturers.* Since it is a deposit method, it would be expected to serve as a good measure of the tendency of the smoke to clog flues and chimneys.

In the *I.C.H.A.M.* method, a specially designed clean pyrex glass rod about 5/16 in. in diameter by four in. in length is inserted into a horizontal portion of the flue through a small hole in the top. The rod is thus in a vertical position, normal to the flow of smoke. The rod is exposed to the smoke for a total of 20 min. For the purpose of obtaining a uniform deposit, the rod is rotated 180 deg about its longitudinal axis when half of the exposure time has elapsed.

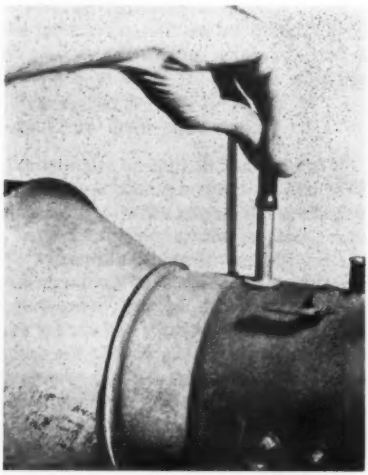
The rod is removed from the stack at the end of the 20 min and the extent of deposit is determined by measuring the light which will pass through the coated rod on a diameter. This is done by inserting the rod in an instrument containing a light source, photocell and milliammeter, housed in a metal cabinet.

Step 1: Clean smoke rod thoroughly.

Step 2: Insert clean rod in meter and rotate to point of lowest reading. Adjust meter to read 100 transmission. Remove rod and note the meter reading on air.



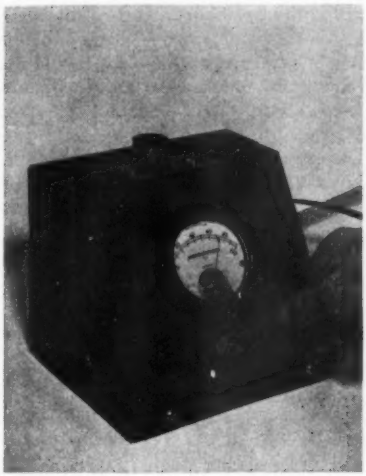
Step 3: Insert rod in stack opening and start timing simultaneously.



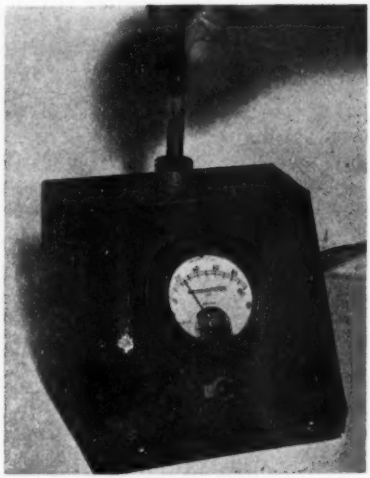
Step 4: Rotate smoke rod 180 deg in stack when exactly half the exposure time has elapsed.

(See opposite page for Steps 5 and 6)

FIG. 3. STEPS IN OBTAINING *I.C.H.A.M.* SMOKE VALUES



Step 5: Adjust meter to read the same light transmission through air as observed in Step 2.



Step 6: Insert exposed smoke rod in meter and rotate. Lowest observed reading is the *I.C.H.A.M.* smoke.

FIG. 3. *(continued)* STEPS IN OBTAINING *I.C.H.A.M.* SMOKE VALUES

The instrument is provided with a support for the glass rod and an arrangement for directing a beam of light through the rod onto the photocell. The cell is connected to a meter containing a scale graduated from 0 to 100. The voltage to the light source is maintained constant by means of a constant voltage transformer. The light meter must be set at the beginning of each test run by adjusting the light intensity so that the meter indicates 100 with the clean glass rod inserted in it. In order to reproduce this setting when ready to make measurements on the coated rod and before the rod is inserted, the meter is set to the same value of light transmission as was noted immediately after the clean rod had been removed when the meter was adjusted for 100.

When the meter has been properly adjusted, the coated rod is inserted and rotated slowly. Because of variations in the deposit and in the rod, fluctuations of meter reading of several percent will be noted. The lowest value observed is taken as the reading. Since similar fluctuations are observed when rotating a clean rod, the initial setting is made at 100 with the clean rod in the position of lowest light transmission.

The various steps in the procedure are shown in the series of photographs in Fig. 3.

Since the readings are based on light transmitted, the amount of deposit or smoke is proportional to 100 minus the *I.C.H.A.M.* reading. Values of *I.C.H.A.M.* smoke are usually reported as meter readings and hence increase with decreasing smoke.

N.C.S. Smoke Method. The *N.C.S.* meter is the name given to a modification of the filter spot method developed at North Carolina State College. In this apparatus a fixed volume of smoke is drawn through white filter paper and the intensity of the spot produced is compared visually with a series of standard spots. The procedure is essentially a measure of the total amount of dispersed solids in the smoke.

A schematic diagram of the apparatus is given in Fig. 4. This shows the valve arrangements for the three steps in sampling. The water flowing from the 18 liter bottle draws the sample of smoke from the stack through the filter

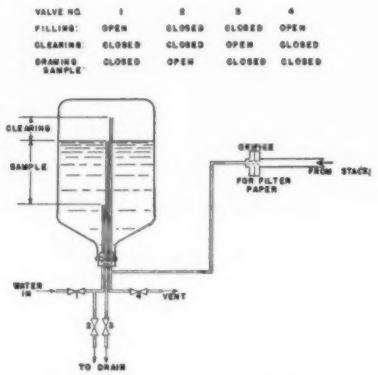


FIG. 4. DIAGRAM OF *N.C.S.* SMOKE APPARATUS WITH OPERATING POSITION OF VALVES. AS SHOWN, THE WATER LEVEL IN THE BOTTLE IS IN POSITION TO START DRAWING A SMOKE SAMPLE ACROSS A CLEAN FILTER DISC

paper into the bottle. The volume of the sample is regulated automatically by the relative positions of the two overflow pipes. As currently operated, a sample of 8200 cc is drawn through the filter paper in approximately 160 sec.

The filter paper is held between two carefully finished brass fittings. A gasket, cut with an inner hole the size of the hole in the brass fittings, is used to prevent leakage of air into the system. The diameter of the spot is about $1\frac{5}{16}$ in. The filter paper used is Whatman No. 4.

Typical spot samples are shown in Fig. 5 for several ranges of smoke. In this photograph, two spots have been made on each filter paper to show the general reproducibility of results. The penciled numbers by each spot correspond to the rating obtained by comparison with the scale of standard spots. The scale used for comparing smoke spots on the filter paper is one developed by The Shell Oil Co. for use with its Shell Smoke Meter, which operates on the same principle. This scale consists of 19 standard spots ranging in density from solid white to solid black. The scale begins numerically at 0 for white and goes

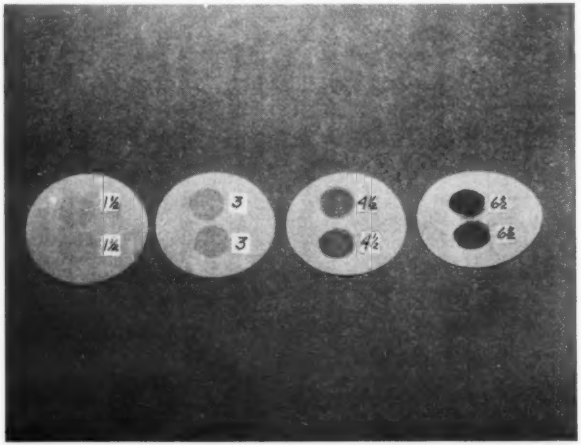


FIG. 5. PHOTOGRAPH OF SMOKE SPOTS OBTAINED USING THE *N.C.S.* SMOKE METER

The smoke spot values shown are from left to right 1½, 3, 4½ and 6½ on the Shell Oil Co. scale. The two spots on each filter disc show the reproducibility of results by this method

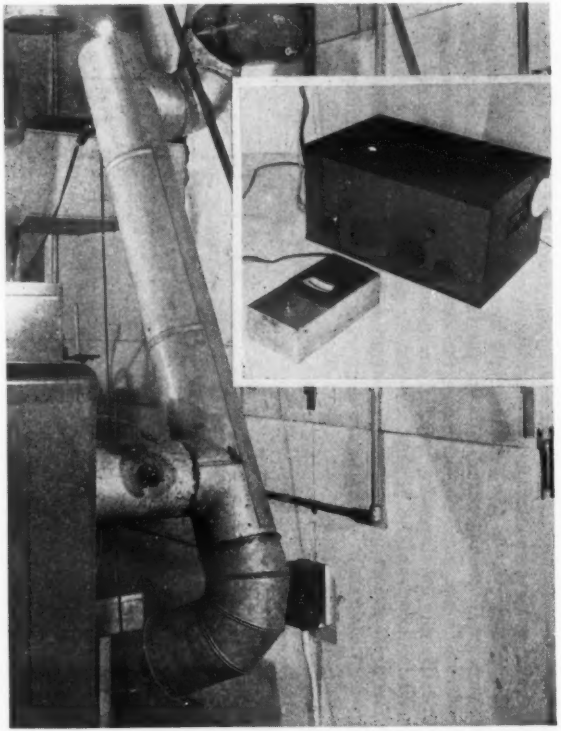


FIG. 6. PHOTOELECTRIC SMOKE INSTALLATION AT NORTH CAROLINA STATE COLLEGE

Photograph of offset stack section with lamp assembly mounted at top left, and *photoelectric* cell mounted at end of elbow, lower left. Insert shows control cabinet for operating two photoelectric installations; microammeter is in lower left of insert

to 9 for black by units of one half, thus giving increasing numerical values for increasing smoke densities.

Photoelectric Smoke Method. The light transmission method was adapted to measure smoke photoelectrically in the Engineering Research Laboratory. As mentioned in the previous discussion, the method is not a direct measure of either quantity of suspended material or its nature. It, therefore, differs distinctly from both the *I.C.H.A.M.* and *N.C.S.* methods.

The arrangement used in this investigation is shown in the photographs of Fig. 6. A diagram of the apparatus is given in Fig. 7. To obtain an appropriate

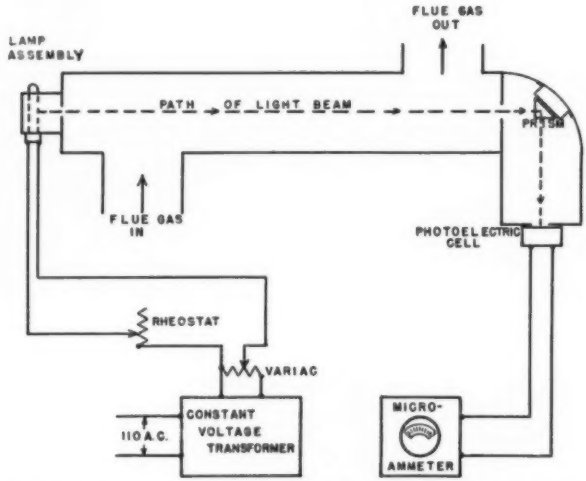


FIG. 7. DIAGRAM OF *Photoelectric* SMOKE APPARATUS WITH ESSENTIAL PARTS AS INSTALLED AT NORTH CAROLINA STATE COLLEGE

length of path, the flue gas is passed through an offset section of the stack four feet long. A 100 watt projection lamp is mounted at one end of this section of pipe. The output of a constant voltage transformer supplies the current for this lamp through a variable transformer in parallel and a rheostat in series with the lamp. Light from the lamp source enters the flue gas along the axis of the stack through a two-inch diameter hole in a cap over the end of the pipe.

An elbow is mounted at the opposite end of this section of stack with a prism set at the center of the bend in such a way that light from the projection lamp is deflected onto a *photoelectric* cell. The photoelectric cell is mounted at the end of the elbow rather than at the end of the four foot section of pipe in order to remove it from the effects of heat from the stack gases.

The output of the *photoelectric* cell is indicated by means of a microammeter. Before starting a test with the furnace, the light intensity from the lamp is adjusted so that some predetermined and convenient value is obtained on the

microammeter. It is best to make this initial setting on a preheated stack. Care must be taken to assure that no products of combustion are present when this setting is made.

Readings taken while smoke is being produced will be less than the initial setting since the smoke interferes with passage of the beam of light, and the output of the *photoelectric* cell will be accordingly less. Thus, the difference between the initial microammeter reading and the test reading or microampere drop is a measure of the quantity or density of the smoke passing through the smoke section.

Smoke readings are reported in terms of this microampere drop and hence, increase in value with increasing smoke densities.

COMPARISON OF SMOKE METHODS

The extensive tests being carried out on the effect of combustion conditions on the burning of fuel oils afforded an opportunity for comparing the results

TABLE 1—COMPARISON OF EFFECT OF EXCESS AIR ON VALUES OF *I.C.H.A.M.*, *N.C.S.*, AND *Photoelectric* SMOKE READINGS

(All data obtained burning light fuel oil in a forced draft pot-type burner)

Excess Air.....%	27.5	23.1	17.8	16.5
Fuel Rate.....lb/hr	4.76	4.84	4.88	4.90
I.C.H.A.M. Smoke Reading.....	78	73	52	32
N.C.S. Smoke (Shell Co. Units)..	2.0	2.5	4.0	5.0
Photoelectric Smoke (milli-amp drop).....	0.3	0.6	0.8	1.8
Stack Temperature.....Fahr	782	780	765	767
Room Temperature.....Fahr	76	77	78	82

obtained with the three smoke meters under a variety of conditions. In the standard tests being run the fuel rate and the air rate were varied for several types of domestic heating units when burning a number of different fuel oils. The data presented here, however, are limited to a single type of furnace using a forced draft, pot type unit.

An increase of the amount of excess air used in combustion usually reduces the amount of smoke produced. The data in Table 1 show the effect of excess air on the smoke values obtained by each of the three methods when the fuel rate is held constant. It will be noted that although the *photoelectric* smoke value increased six fold when the percent excess air was reduced from 27.5 to 16.5, the *N.C.S.* smoke only increased two and one half times. Comparison of these three methods indicates that although all show increasing smoke with decreasing excess air, the relationship among the three is by no means linear.

There are several reasons for this. In the first place each of the scales is arbitrary and cannot be said to be a linear measure of smoke density. Consequently the relationship between them will not be linear. This, however, would not preclude a direct correlation between any two sets of values.

The second and more important reason for the lack of agreement between methods is that other characteristics of the smoke change along with its density.

TABLE 2—SUMMARY OF SMOKE DATA FROM THREE METERS

(All data obtained in tests of a forced-draft pot-type burner)

Group A—High Fuel Rates					*Group B—Intermediate Fuel Rates*					*Group C—Low Fuel Rates*				
Fuel	Smoke Meter Reading				Fuel	Smoke Meter Reading				Fuel	Smoke Meter Reading			
Rate Lb/Hr	*I.C.H.A.M.*	(100—*I.C.H.A.M.*)	*NCS*	Photo-Electric	Rate Lb/Hr	*I.C.H.A.M.*	(100—*I.C.H.A.M.*)	*NCS*	Photo-Electric	Rate Lb/Hr	*I.C.H.A.M.*	(100—*I.C.H.A.M.*)	*NCS*	Photo-Electric
4.46	95	5	0.5	0.2	4.14	40	60	4.7	1.7	3.67	55	45	—	1.5
4.44	82	18	2.2	0.6	4.13	65	35	4.3	1.0	3.70	42	58	—	2.2
4.45	16	84	6.2	2.6	4.12	80	20	3.7	0.4	3.64	25	75	—	3.0
4.54	36	64	4.3	1.5	4.17	73	27	—	0.8	3.48	51	49	4.5	1.7
4.56	83	17	2.2	0.5	4.10	56	44	—	1.7	3.48	81	19	3.0	1.0
4.58	50	50	4.0	1.2	4.06	37	63	—	2.2	3.52	94	6	1.5	0.6
4.39	73	27	—	0.9	4.12	86	14	1.7	0.4	3.36	90	10	2.0	0.6
4.52	50	50	—	1.6	4.03	64	36	3.7	1.2	3.22	48	52	5.0	2.9
4.67	63	37	—	1.3	4.02	35	65	5.5	2.3	3.27	21	79	7.5	5.5
4.60	11	89	—	3.4	3.94	79	21	2.7	0.2	3.24	95	5	1.3	0.2
4.63	48	52	—	1.4	3.96	64	36	3.7	0.8	3.14	81	19	3.1	0.7
4.65	25	75	—	2.3	4.06	41	59	5.0	1.5	3.29	32	68	6.0	2.8
4.65	91	9	2.2	0.4	3.84	68	32	—	1.2	2.97	69	31	—	1.7
4.70	66	34	4.0	1.2	3.80	18	82	—	3.7	3.00	37	63	—	3.3
4.71	19	81	6.2	3.8	3.93	66	34	—	1.0	2.84	44	56	—	2.3
4.84	73	27	2.5	0.6	3.94	22	78	—	2.8	2.93	18	82	—	4.9
4.76	78	22	2.0	0.3										
4.88	52	48	4.0	0.8										
4.90	32	68	5.0	1.8										
4.42	91	9	1.2	0.2										
4.50	71	29	3.0	0.8										
4.50	27	73	5.0	2.2										
4.15	94	6	1.2	0.4										
4.24	51	49	4.3	1.6										
4.17	57	43	4.3	1.1										
4.23	39	61	5.1	2.0										

Reducing the amount of excess air alters the nature (size, shape and composition) of the smoke particles as well as their concentration.

This change in the nature of the smoke particles is even more pronounced with variations in the fuel burning rate of the burner. To illustrate this, data are given in Table 2, representing average results for a number of runs at various fuel rates. The data are listed in order of decreasing fuel rate and have been arbitrarily grouped as high, intermediate and low fuel rates.

Comparisons of results are presented graphically in Figs. 8, 9, 10 and 11. In Fig. 8 the relationship between *I.C.H.A.M.* smoke and *photoelectric* smoke values is given. To avoid the natural inverse relationship, values of 100 minus *I.C.H.A.M.* readings have been used. It will be observed that there is considerable spread of the data and this is particularly pronounced at high smoke

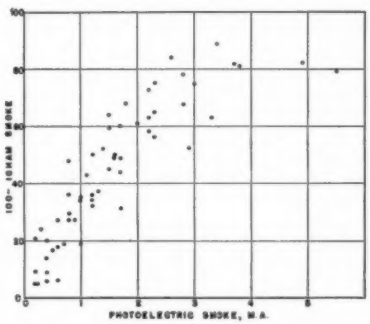


FIG. 8. SUMMARY OF DATA SHOWING RELATION OF *I.C.H.A.M.* TO *Photoelectric* SMOKE INDICATION

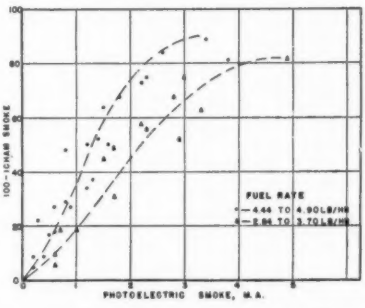


FIG. 9. EFFECT OF CHANGE IN FUEL RATE ON RELATION BETWEEN *I.C.H.A.M.* AND *Photoelectric* SMOKE INDICATION

density values on the *I.C.H.A.M.* scale. This indicates that some smokes having high deposit forming tendencies are much more transparent than others.

Only the data of Groups A and B are plotted in Fig. 9. Here the data do fall fairly distinctly into the same groups on the graph. For the forced draft, pot type burner, it appears that at low fuel rates, the smoke is less transparent for a given amount of deposit than at high fuel rates. This possibly indicates a higher degree of dispersion or a lower average particle size at the low fuel rates. Another explanation is that at high fuel rates, the particles are more adhesive and less carbonized, and hence have a greater tendency to deposit than particles produced at low fuel rates. Actually it seems probable that both factors are operating simultaneously. The changes in the stack temperature with changes in the fuel rate, as indicative of variations in combustion conditions, substantiate this idea.

The important point to be drawn from Fig. 9 is that no reliable indication of the soot depositing tendency of a smoke may be drawn from the *photoelectric* reading. A wide variation in fuel rate at a given *photoelectric* smoke reading

may have as much effect on *I.C.H.A.M.* smoke as does a wide variation in *photoelectric* smoke at a constant fuel rate.

The comparison of *N.C.S.* smoke values with *I.C.H.A.M.* smoke is shown in Fig. 10. The agreement here is much better and more nearly linear than the previous values. It will be observed, however, that an average line drawn through the plotted points will intercept the *N.C.S.* scale at one instead of zero. The *N.C.S.* filter spot tends to give visible readings even though the smoke particles have little tendency to deposit as shown by the *I.C.H.A.M.* test. It seems probable that a shift in this relative sensitivity at low smoke values could

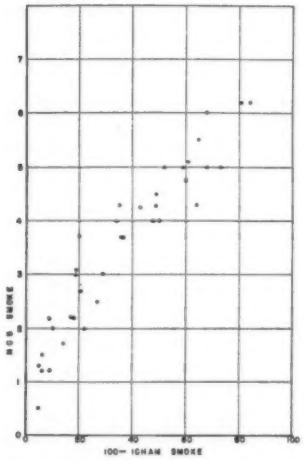


FIG. 10. SUMMARY OF DATA SHOWING RELATION OF *N.C.S.* TO *I.C.H.A.M.* SMOKE INDICATION

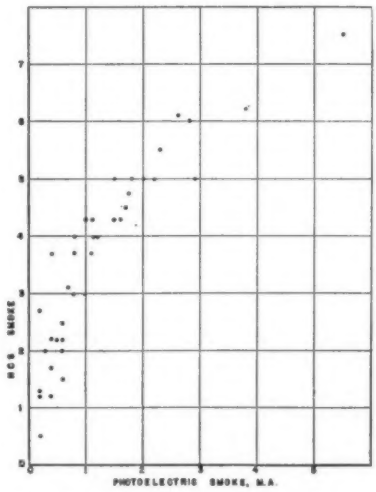


FIG. 11. SUMMARY OF DATA SHOWING RELATION OF *N.C.S.* TO *Photoelectric* SMOKE INDICATION

be made by altering the time of exposure for the *I.C.H.A.M.* test or the size of the sample drawn in the *N.C.S.* test. If this were done, the two methods might be brought into sufficiently good agreement to establish a correlation or rough conversion factor between them. It should be remembered that all of these values were taken with smoke from a forced draft, pot type oil burner. Entirely different relationships might be obtained with coal- or wood-fired smoke or with other types of furnaces.

The relation between *photoelectric* smoke and *N.C.S.* smoke values is shown in Fig. 11. The data fall on a more or less parabolic curve with the most severe scattering being noticed at the low smoke values. It will be observed that smokes giving quite distinct *N.C.S.* spots as high as four may show very low *photoelectric* smoke values. A few relatively large particles filtered out will give a heavy spot test, but since light is scattered more by numbers than size, the transmission will still be high.

CONCLUSIONS

Each of the graphs just presented shows that several smoke samples, each of the same value by one method, may have a variety of values when measured by some other procedure. Conversely, smoke samples which appear distinctly different when measured by one procedure, say the *N.C.S.* method, may be indistinguishable by a procedure such as the *photoelectric.*

It is, therefore, impossible to say that one method is better than the other. It can only be said that they are different. If no correlation exists when all measurements are made on fuel oil burned in a single furnace it is obvious that agreement could not possibly be expected with a variety of furnaces or different fuels.

The conclusion must be drawn that each method measures different aspects of smoke and not smoke itself. The simultaneous use of several methods gives more complete information than any one alone can give. Use of several methods should aid in prediction of the effect of combustion conditions on the performance of fuels.

ACKNOWLEDGMENT

The authors wish to express their appreciation for the support of The Texas Co. in making this work possible.

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DISCUSSION

A. J. JOHNSON, Lansdowne, Pa. (WRITTEN): Beatty and Deas have presented an outstanding evaluation of the several accepted methods of measuring smoke. Of the several such methods either suggested or proposed, the Ringelmann chart and the

Umbrascope are the only ones that are sufficiently portable for the necessarily quick and unannounced inspections of municipal authorities. Nevertheless, both are notoriously inadequate for several reasons even aside from considerations of possible inaccuracy.

The most important limitation of visual inspection of any type is that its use is restricted to daylight hours and clear weather thus leaving a city seeking cleanliness without any protection whatsoever for over half of the winter hours. Not only are these the very hours when the most fuel may be used but, in addition, many plants take advantage to perform soot burning and blowing operations at night. In fact, I well remember an experience as a young engineer in a city that is particularly boastful of its cleanliness and fuel restrictions. One of my duties was to return to the plant on one moonless night a month to supervise burning out the waste-heat boilers. I have often wondered whether nearby residents ever traced the source of the hunks of soot which they invariably found on their doorsteps in the morning.

Even in the absence of any such premeditated increase in pollution at night, the firing rate of heating equipment is normally stepped up to several times its daytime rate near sunset. Smoke and fumes are thus materially increased at the very time when they cannot be checked by the legally defined method. This is particularly true in residential neighborhoods where smoke is the most objectionable. Industrial plants likewise become careless at night and usually have less experienced supervision.

A further marked weakness of visual inspection is the prohibitive expense of attempting to place enough men in the field to effect anything but the spottiest kind of coverage.

From the air pollution angle, the ideal solution would seem to be to require installation of recording smoke meters of the best available type, possibly photoelectric, on all plants where the equipment, fuel, or combination of the two has been ruled a potential smoke producer. The resultant charts would then be subject to municipal inspection. The accuracy of quantitative measurements would not be too important as long as all plants in an area used meters of the same type so as to insure relative accuracy.

In answer to the thought that the compulsory addition of recording smoke meters would involve prohibitive expense or inconvenience; this would be infinitely less of a hardship than the present tendency to force conversion to rather arbitrarily defined fuels and equipment. In fact plants would then have the option of installing city inspected recorders or of converting to combustion conditions considered adequately free from smoke potentialities. In most cases, the expense of such recorders could be justified by subsequent fuel savings.

Such a plan would eliminate the present serious weakness of many present ordinances in that there would no longer be any need for exempting large, and frequently objectionable, utilities and mills on the ground that equipment for fuel change would be impossible or prohibitively expensive. Such plants would merely be required to install recorders and to be subject to their readings.

P. R. ACHENBACH, Washington, D. C. (WRITTEN): The authors are to be complimented for an interesting and worthwhile comparison of the several methods of smoke measurement that are in current use. While the present study does not logically lead to a choice of one method as being superior to the others, it is to be hoped that further correlation of the smoke indications with other characteristics of the smoke would indicate the best method and lead to the establishment of limits of smoke production that would be related to the degree of nuisance involved.

The first necessity in determining the characteristics of a fluid flowing through a pipe such as a smokepipe is that of obtaining a representative sample for analysis. Of the three methods discussed in this paper, the *I.C.H.A.M.* method is probably the one most to be questioned with respect to this requirement.

I would like to ask the author whether any study was made to determine if the glass rod collects proportionate amounts of all sizes of particles flowing through the smokepipe. In Fig. 3, Step 4, of the paper, was the flue gas flowing to the right or to the left? The smoke rod is located near a place where the smokepipe changes diameter which might cause eddying in the gas stream and affect the smoke deposit.

Then, with regard to the *N.C.S.* method, human judgment is required in comparing the deposit with a standard spot. I believe, human judgment is not involved in the measurements of the deposits or the smoke in the other two methods.

The nuisance characteristics of smoke depend primarily on its blackness when considering the discoloration of interior furnishings, wearing apparel and merchandise or the discoloration of the exterior surfaces of buildings, although particle size determines how far it will be distributed from a given source. However, once the decision has been made to remove the smoke or reduce the amount of smoke by filtering methods, the mass of solid matter to be removed per unit volume of outdoor air and the distribution of particle size become more important than the color. Therefore, the heating and air conditioning engineer is primarily concerned with the properties of smoke that determine how readily it can be filtered from the air.

The insulating effect of smoke or soot on the heat transfer surfaces in heating equipment is probably of greater importance in lowering the efficiency of furnaces and boilers than is the incomplete combustion loss sometimes associated with smoke and inadequate air supply. Studies carried out at the National Bureau of Standards several years ago showed that a layer of soot approximately $\frac{1}{32}$ in. thick reduced the efficiency of an oil-fired boiler about 5 percent because of the insulating effect of the soot. The deposition of soot in a heating plant probably depends partly on the mass of soot per unit volume of gases, partly on particle size, possibly on the tarry characteristics of the soot, and perhaps other factors.

If tests such as those summarized in Tables 1 and 2 in the present paper were carried out simultaneously with analyses of particle size, mass of solid matter per unit volume of gases, and volatile content of the soot, some correlation between these physical characteristics and the smoke measurements obtained by the three methods used might be revealed that would show one of the methods of measurement to be better than the others. Some relationship between the tendency of soot to deposit on heating surfaces and the physical characteristics of the soot might also be discovered along with a good index for measuring this tendency. It is noted in Table 2 that for the Group A tests the smoke index values were from 12 to 17 times as great for some tests as for others using about the same fuel rate. Somewhat similar results are shown for Group B and Group C tests. Can these variations be attributed to differences in draft level or to some other conditions of the tests?

This investigation presents a much needed comparison between the several smoke-indicating methods and points the way to further research on this subject that is of importance to every city and every heating and air conditioning engineer.

H. G. DYKTOR, Cleveland, Ohio (WRITTEN): I commend Professor Beatty on this excellent paper and on his realistic conclusions. Measurement of emitted smoke is the main interest of an official enforcement agency engaged in the reduction of air pollution to a tolerable level rather than in the other phases discussed by Professor Beatty. The measurements of deposit of soot in flues and stacks and of the efficiency of combustion of fuel are of subordinate interest in that they affect indirectly the emission rates of smoke and fly ash. Because of long usage, this city has adopted the Ringelmann chart as a measure of smoke emission, and it works out satisfactorily from our particular angle in spite of its many shortcomings. It is only when violations occur that our staff investigates the fuel, the equipment, and the operation because it is obvious then that one of these three factors is unsatisfactory.

I agree with Professor Beatty that no matter which method is used for measuring smoke, results are reported in arbitrary units so that they are valid only for compara-

tive purposes and do not define either the quantity of material emitted nor the damage done by that material. Much more work remains to be done in this respect. In the meantime, through education (at both the manufacturing and operating levels) and annual inspection by competent inspectors, the problems of proper combustion are being solved, one by one.

R. C. CROSS, Chicago, Ill.: The matter of smoke measurement is a very complex one, but I think the work done by the author is rather helpful to those engaged in the laboratory phases of such work.

I recall doing some work of a similar nature several years ago with the photoelectric method. At that time we were trying to correlate it with some field work which was to be done, and we attempted to plot our readings against the Ringelmann chart. It is interesting to note that we, too, obtained a shotgun pattern similar to the authors' graphs. Even so, we were able to draw a reasonable correlated curve between the two methods.

I think it would be most interesting if the results that the author obtained could be compared to the Ringelmann method, because, despite its very obvious shortcomings, some of which have been mentioned by previous commentators, it does remain the method on which most smoke regulatory bodies base their opinions.

E. K. CAMPBELL, Kansas City, Mo.: The difficulties set out by the author were presented to a committee of engineers composed of members of the Kansas City Chapter attempting to guide the authorities in Kansas City to a better smoke ordinance. They adopted some of our ideas resulting in some improvement.

However, when we attempted to write into the ordinance specifications regarding the fly ash or dust that is carried out with the smoke, there was no information that was worth anything from a practical standpoint. One of the members of the committee represented a firm that sold a dust collector to one of the power plants there. The contract for the installation of this dust collector required them to put on a test to demonstrate the efficiency of the dust collector. They figured $1,000 for the cost of that test.

Municipal authorities obviously cannot make tests of that sort, but from the standpoint of the protection of the public we all agreed that the matter of the fly ash, in communities where large amounts of soft coal are burned, is probably more important than the visible smoke, but so far there has been no standard test available. We had all the information at the hands of the *A.S.M.E.* Smoke Committee and there was nothing that could be set up. So we were forced to use the nuisance value of the dust which is scattered over the areas surrounding the different plants as the criterion for setting up rules.

G. V. PARMELEE, Cleveland, Ohio: Professor Beatty has shown in his paper that he has had some of the same difficulties which we have had at the A.S.H.V.E. Research Laboratory in smoke measurements. In addition to those problems which he has discussed, we have found that the fuel itself has a bearing on the smoke measurement. This is Fig. 5 from the A.S.H.V.E. Research paper, Smoke Measurement in a Fuel Oil Test Unit, which is being presented at this session by title only. It shows the results of tests on three different fuel oils in which two methods of evaluating smoke density were used. One is the photoelectric method, which is plotted as ordinate in the figure. The ordinate is the reduction in the microampere output of the photoelectric cell. The abscissa gives smoke density in terms of the reflectance of filter paper through which the smoke sample has been passed. You will see that for a given smoke spot reflectance, say of 20, there is quite a range in the microampere output of the photoelectric cell. Conversely, for a given microampere reduction in output there is a sizeable variation in the reflectance of the filter paper. This appears to be due to particle size distribution and the number of particles. The figure shows that this is just one more matter that has to be taken into consideration.

I would like to ask Professor Beatty if he experienced any difficulty in obtaining steady output from his photoelectric cell. We had trouble so that in our final tests, we used the reflectance method for obtaining smoke density. Reflectances were evaluated by a photoelectric reflectance meter.

M. F. DIXON, Toronto, Ont., Can.: I would like to ask Professor Beatty one or two questions concerning the measurement or the evaluation of white smoke. I would imagine that in the testing he did in a pot-type burner with just gravity air he probably did not produce any white smoke, but what I am referring to is the type of white smoke that you have with a very great quantity of excess air.

I must say I know personally very little about it except I have often seen it. I am wondering in the case of white smoke—I imagine you would get very low photoelectric cell readings, but if you actually test that white smoke by collecting it on a filter pad do you have a combination of excess air and also incomplete combustion, and is there considerable sediment in it?

AUTHORS' CLOSURE: The authors wish to thank the discussers for their discussions and additions to this paper. I should like first to answer some of the more specific questions raised, and then to comment generally on the question raised by several as to the relationship between smoke measurement values and smoke nuisance.

Mr. Achenbach raises the important question of whether each smoke measurement method used takes a representative sample. The answer is, *probably no*, but since the same setups were used for all tests reported, the relative values are certainly comparable. With regard to the wide variation in smoke values listed in Table 2, it should be pointed out these were obtained deliberately and reflect variations in air rate, fuel rate, and fuel composition.

Mr. Dixon asks about evaluation of white smoke. In general a smoke is white because of fine particle size. Such a smoke is relatively opaque and hence will give high photoelectric smoke values. On the other hand, it may be difficult to remove by filtration or deposition and hence may give low values by such methods as the *I.C.H.A.M.* or *N.C.S.*

Mr. Parmelee asks about our experience with constancy of photoelectric readings. In the tests reported, we had no trouble. Recently we moved the equipment and installed the stack vertically instead of at the 60 deg angle previously used. This new setup did give some trouble due to settling of soot onto the lenses. It seems possible to me that the trouble Mr. Parmelee reports may have been due to the use of a horizontal stack in his work. Our observations agree with Mr. Parmelee's on the effect of fuel composition as an important variable. Our results on a wide variety of furnace oils have not yet revealed any way in which to predict these from any other known properties of the oils.

Most of those commenting on this paper have expressed interest on possible relationship between smoke as measured by one of these three methods and smoke nuisance in a city. It was pointed out by Mr. Campbell and by Mr. Johnson that fly ash and the tendency of soot to settle are important factors. It is our opinion that these factors would not be adequately determined by any of the three methods we used, nor by the Ringelmann method.

Possible relationships between Ringelmann method and photoelectric, *N.C.S.* or *I.C.H.A.M.* smoke do not appear promising. Ringelmann smoke is measured outside the stack after the gas is cooled and has been partially mixed with outside air. Consequently identical smokes within the chimney may give widely different Ringelmann values due to atmospheric conditions. The presence of large amounts of water vapor in the stack gases will increase smoke density on cloudy, cold, damp days and have much less effect on hot, clear, dry days.

The relationship of smoke to smog appears to be related to the presence of transparent gases such as sulfur dioxide, as well as to particles of soot, fly ash, or other

fumes. It thus appears to the authors that intelligent smoke regulation must be preceded by careful stipulation of the objectionable characteristics of smoke, as for example its light obscuring characteristics, its tendency to settle and deposit, and its chemical composition. When these characteristics have been specified, the appropriate method for measuring these can be selected. There seems to be no reason for believing that any single smoke measurement method will ever evaluate all the objectionable characteristics of smoke in their proper relationship.

1412

No. 1412

SMOKE MEASUREMENT IN A FUEL OIL TEST UNIT

By D. W. Locklin* and G. V. Parmelee**, Cleveland, Ohio

This paper is the result of research carried on by The American Society of Heating and Ventilating Engineers, in co-operation with the *Oil-Heat Institute of America, Inc.*, at the A.S.H.V.E. Research Laboratory, located at 7218 Euclid Avenue, Cleveland 3, Ohio.

PRESENT-DAY methods of producing gasoline from crude oil have markedly affected the chemical composition and physical structure of those by-products which remain for use as fuel in oil burners. This is particularly true of the lighter portions which are used in domestic heating units. Whereas some years ago oil burner fuels were the product of straight-run distillation or thermal cracking processes, present-day fuels may contain or consist entirely of products from catalytic cracking plants. Experience has shown that such fuels may have different burning properties than fuels from a straight-run distillation process although both may possess physical properties that place them in the same classification.

To permit the proper selection of fuel for the thousands of oil burners now in use, some method of determining the burning charcteristics of fuel oil is needed. One method which has been proposed for obtaining these characteristics for various oils is to study their combustion in a test unit which closely simulates typical field conditions and to compare the oils on the basis of their observed burning performance. The test unit referred to in this paper is a pressure atomizing type unit designed by the *Oil-Heat Institute of America, Inc.*, through whose Engineering Committee arrangements were made in 1947 with the Committee on Research of The American Society of Heating and Ventilating Engineers for conducting a series of investigations.

A preliminary field investigation was made in December 1947 for the purpose of determining, in a general way, whether the test unit could be used to differentiate between various fuel oils. The principle criterion of fuel performance was

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considered to be the smoke production with respect to excess air supplied. The field observations made showed that the unit had definite possibilities but that improvements in instrumentation and techniques were required if it were to be used as a means of rating fuel oils. Two extensive series of tests were therefore made during 1948 and 1949 at the A.S.H.V.E. Research Laboratory in Cleveland.

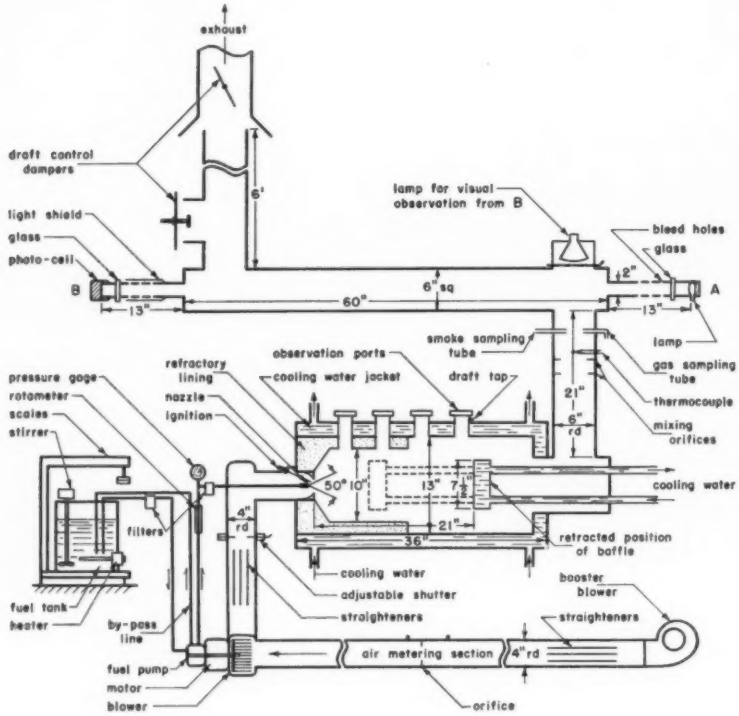

FIG. 1. SCHEMATIC ELEVATION OF TEST UNIT

This paper and a companion paper summarize the investigations and tests mentioned. The scope of this paper is confined to the methods of smoke measurement used with the unit, while the second paper [1] describes the performance of the unit and several different fuel oils which were burned in it.

DESCRIPTION OF TEST UNIT

Fig. 1 shows a schematic diagram of the unit and a portion of its instrumentation. The engineering staff of the A.S.H.V.E. Research Laboratory made

[1] Exponent numerals refer to References.

certain changes during the course of this study and completed most of the instrumentation. A photograph of the unit as set up for the second series of tests is shown in Fig. 2.

Basically the unit consists of a pressure-atomizing-type burner firing axially into a cylindrical water-jacketed combustion chamber, a portion of which is lined with refractory material as shown. The nozzle has a nominal capacity of 1 gph and produces a hollow-cone spray. Flame length was determined by noting the position of the movable water-cooled baffle at which the flame tips just reached the plane of the baffle surface. Air supply and combustion chamber draft were controlled by adjusting (1) the cold air inlet damper located in the

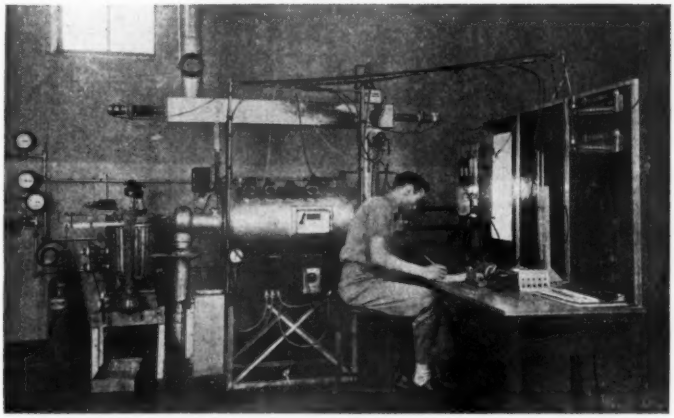

FIG. 2. THE COMBUSTION TEST UNIT DURING TESTS

outlet of the smoke observation tube, (2) the butterfly damper in the exhaust flue, and (3) the iris type air supply shutter.

METHODS OF SMOKE MEASUREMENT

Investigators have used many different techniques for smoke measurement, both qualitatively and quantitatively, in considering combustion efficiency and atmospheric pollution. These techniques may be classified as follows:

1. Visual shade comparison of smoke issuing from a stack.

2. Transmission of radiant energy.

A. *Radiant Source*
   - *a.* Chromatic
   - *b.* Monochromatic

B. *Transmission*
   - *a.* Direct
   - *b.* Reflection

C. *Measurement*
   - *a.* Photoelectric effect
   - *b.* Thermal-electric effect (bolometric)

D. *Sampling*
   - *a.* Measurement in stack on effluent gases
   - *b.* Measurement on sample withdrawn from stack

3. Deposit

A. *Collection*

a. Filtration
b. Sedimentation
c. Impingement
d. Inertial and centrifugal
e. Washing
f. Precipitation: (*a*) thermal
(*b*) electrostatic

B. *Evaluation*

a. Weight
b. Chemical analysis
c. Light transmission
d. Light reflectance
e. Microscopic particle size count
f. Visual shade comparison

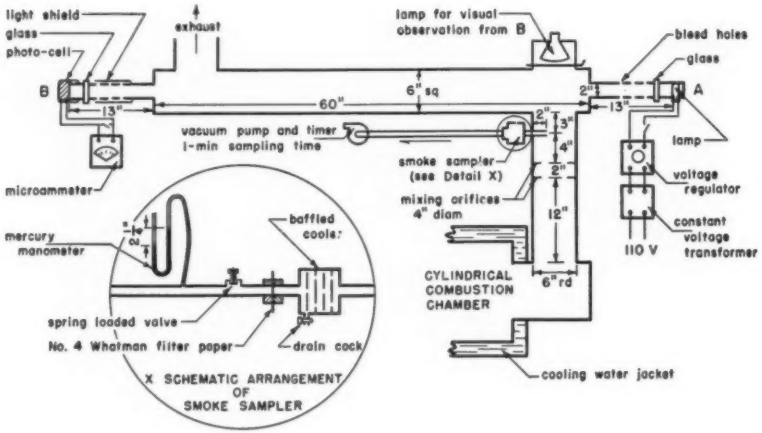

FIG. 3. SCHEMATIC DIAGRAM OF SMOKE MEASURING EQUIPMENT FOR TEST UNIT

These techniques are discussed in articles which are listed chronologically in the Bibliography.

METHODS USED

Direct Light Transmission Measurement in Stack

The unit, as originally designed, incorporated a smoke indicating device patterned after that used by the *Underwriters Laboratories, Inc.*[2] A photronic cell indicated the reduction, due to smoke, of chromatic light transmission through a horizontal breeching 5 ft long, as shown in Fig. 3.

A 100-watt projection lamp and a photronic cell were mounted at opposite ends of the breeching behind protective glass slides. Difficulties in keeping these glasses free of soot prompted the addition of 9-in. extensions of 2-in. copper tubing between the breeching and glasses. Holes drilled in these extensions permitted air to be bled into the system for the purpose of clearing the tube of eddying smoke and keeping the photocell cool. The system was *zeroed* with a

clear but hot stack by adjusting the lamp voltage so that the output of the photronic cell was 8.0 microamperes. The light absoption due to smoke and flue gases in the breeching during operation was indicated by the reduction in the photronic cell output.

Experience in the first series of tests showed that this method of measurement lacked the desired stability and sensitivity at light smoke densities. A supplementary smoke indicating device was therefore investigated for the second series

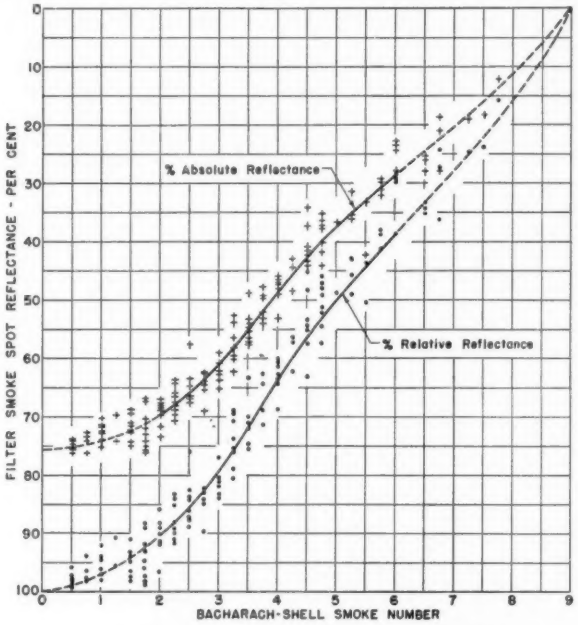

FIG. 4. SMOKE SPOT EVALUATION—REFLECTION METER *vs.* VISUAL

of tests in order to improve reproducibility and range; these characteristics were achieved by the choice of the *filtration* method.

Filtration Method

In this method of smoke indication, a cooled sample of flue gas was drawn through a filter paper for one-minute period and the deposition of soot particles on the filter paper evaluated as an indication of the smoke. This system is schematically shown in Fig. 3. A spring loaded relief valve maintained the pressure differential across the filter paper constant at an indicated suction pressure of 2¼ in. Hg. The resistance of the No. 4 Whatman filter paper used was such that the spot area of 0.216 sq in. would pass approximately 10,000 cc of gas sample per minute at this differential pressure.

In the second series of tests the smoke spot obtained on the filter paper was evaluated by visual comparison with a 9-shade Bacharach-Shell scale [3]. This procedure had certain shortcomings for the purpose of these tests because of the varying effects of particle size, color, light transmission, surface properties, and the human error in judgment. The lack of uniformity of different shade scales also made it difficult to compare results for smoke spots which have been evaluated by different scales. It was therefore decided to use photometric means of evaluating the smoke spots for the third series of tests; the light reflectance of the filter smoke spots was measured by a photoelectric reflection meter, with a green filter [3].

The reflectances of three arbitrary standards were determined by using the absolute reflectance of a block of magnesium carbonate as reference. In evaluating smoke spots the reflection meter scale was adjusted with these standards to read percent absolute reflectance directly. The absolute reflectance of a clean portion of each filter was determined in order that the reflectance of the smoke spot could be expressed relative to that of the clean filter paper. *Percent relative reflectance* is then the percent absolute reflectance of the smoke spot expressed as a percentage of the percent absolute reflectance of the clean filter paper.

Fig. 4 shows the relationship obtained between smoke spot reflectance and the Bacharach-Shell visual shade scale.* Reflectance readings of a given surface were considered reproducible to ± 0.2 percent absolute reflectance; therefore differences in readings of smoke spots taken under the same conditions were due to factors other than the reflectance measurement, such as unsteady combustion and nonuniform sampling. Variations from the averaged smoke spot evaluations at a given setting were generally within ± 2 percent relative reflectance for the lighter oils investigated and within ± 5 percent for the heavier oils.[1]

Characteristics of Light Transmission and Filtration Methods

The optical density of smoke indicated by the light transmission method is given by the expression [4]

$$D = \log_{10} I_0/I \quad \ldots \quad (1)$$

where

D = optical density indicated
I_o = intensity of incident light
I = intensity of transmitted light

Beer's law for light absorption is

$$I_o/I = e^{k_1 nLd} \quad \ldots \quad (2)$$

where

e = base of natural logarithms
k = a constant
n = concentration of smoke in particles per unit volume of gas
L = length of the smoke column
d = mean diameter of the particles

Combining Equations 1 and 2 gives

$$D = k_2 n L d \quad \ldots \quad (3)$$

For a given smoke meter the optical density indicated becomes

$$D = k_3 (n d) \quad \ldots \quad (4)$$

* The Bacharach-Shell scale has been defined as a linear relationship with zero and ten smoke number corresponding respectively to 100 and zero percent relative reflectance.

Thus smoke measurement by light transmission methods is affected by both concentration and mean size of smoke particles.

In the operation of the unit it was found that the relation between the light transmission and the filtration methods was not well defined due to scatter of data; it was apparent, however, that the relation for each oil was different. Smoke data by the two methods are shown in Fig. 5 for three representative oils. It is seen that the light transmission method which was used was not

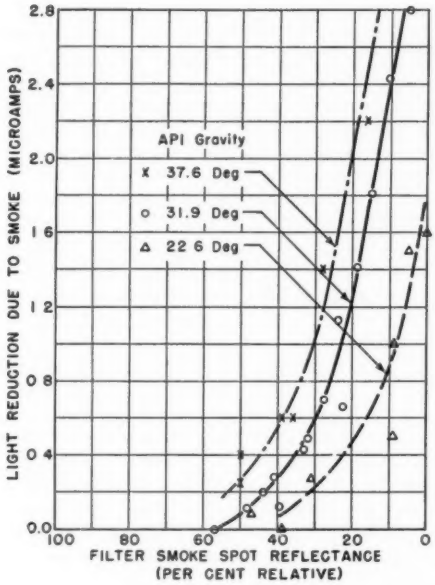

FIG. 5. RELATION OF SMOKE MEASUREMENT BY LIGHT TRANSMISSION AND FILTRATION METHODS

successful in detecting light smoke, although it is recognized that an improvement might be made by refinement of the photronic circuit.

By simultaneously taking filter paper smoke spots and sighting through the breeching from location B of Fig. 3, with illumination from the upper lamp, it was observed that smoke spot reflectances corresponding to the *just visible* smoke were quite different for different oils. This was believed due to differences in particle sizes of the smoke. A portion of this effect could be attributed to sampling characteristics of the filtration system.

The efficiency of sampling was not investigated in detail; however, certain factors are recognized which may have an effect upon sampling. In order to maintain the simplicity of the smoke indicating methods for field use, the smoke sampling tube was inserted into the stack as indicated in Fig. 3 and the sample

was drawn at right angles to the flue gas flow. Variations in the particle size of the smoke, and variations in stack velocity, could conceivably affect the results when sampling in this way. The mean gas velocity in the sampling tube was roughly half that in the stack, though it has been shown [5,6] for sampling tubes directed upstream that the sampling velocity must be equal to the stack velocity in order to insure a representative sample. When the filtration method was used with very dense smoke it was necessary between readings to purge the sampling system of residual soot by drawing a sample of room air through it at high velocity.

It was evident by inspecting filter smoke spots that the particle sizes of the soot deposit were greater for heavier oils than for the lighter oils. The photometric reflectance evaluation of the spots did not completely obviate particle size effects since an identical reflectance reading could be obtained for a large concentration of small particles or a lower concentration of larger particles. Such a characteristic is also inherent in the direct light transmission method. Particle size and shape may have a practical bearing upon aerodynamic suspension of particles in the gas stream, and therefore upon soot deposit on various surfaces.

One feature of the filtration method which was considered desirable for this investigation was the integration of the smoke over a period of time. Although tests were made under steady combustion conditions there were times when intermittent, and sometimes audible, pulsations made readings difficult by the light transmission method. The instantaneous feature of the latter method, however, makes it valuable in certain applications.

In a recent paper [7] reporting the simultaneous measurement of smoke by three methods, the *ICHAM* deposit, direct light transmission, and filtration, it was concluded that all of these are indicative of different aspects of smoke, such as gumminess and particle size, rather than the smoke quantity itself; thus the methods cannot be directly correlated for different conditions.

Methods of smoke measurement should be investigated in order to determine their relation to a weight concentration evaluation. In addition to weight of smoke production, it may be important to consider other aspects of the smoke and their effects upon combustion efficiency, heat transfer, and sooting of flue passages.

CONCLUSIONS

1. Smoke indication by the filtration and light transmission methods did not bear a consistent relationship.

2. Of the two techniques used, the filtration method, with evaluation by photometric reflectance, was the most suitable for use with the test unit. This choice was due to the following considerations:

a. This method possessed greater sensitivity and range in the region of light smoke.

b. A consistent relation was established between visual and photometric reflectance evaluation of filter smoke spots.

3. Further investigation is required on the effects of different sampling techniques.

4. Various aspects of the smoke production which are indicated by different methods should be investigated in relation to weight concentration of smoke and their effects upon appliance performance.

ACKNOWLEDGMENT

The authors acknowledge with thanks the cooperation and suggestions received from their colleagues of the A.S.H.V.E. Research Laboratory staff and members of the Engineering Committee of the *Oil-Heat Institute of America, Inc.*

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1413

No. 1413

RATING OF FUEL OILS BY A TEST UNIT

By D. W. LOCKLIN* AND G. V. PARMELEE**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in co-operation with the *Oil-Heat Institute of America, Inc.*, at the A.S.H.V.E. Research Laboratory, located at 7218 Euclid Avenue, Cleveland 3, Ohio.

A PROPOSED method of measuring the burning characteristics of fuel oils is to study their combustion in a test unit which closely simulates typical field conditions and to compare oils on the basis of their observed burning performance. A test unit designed and constructed for this purpose by the *Oil-Heat Institute of America, Inc.*, was studied by the A.S.H.V.E. Research Laboratory and is the subject of this paper. A previous paper [1] described the methods used in determining smoke density of the combustion products. This paper describes briefly the unit, the standardized test procedure used, and the results of tests on six different oils. The oils are rated on the basis of their smoking tendencies at given rates of air supply. The ability to obtain reproducible results by means of the test unit was determined and the influence of certain operating conditions on the burning qualities of a given fuel was investigated. A long range view of the uses of this unit includes the establishing of a simple test or index which can be used in the field to determine the burning quality of fuel oils.

DESCRIPTION OF TEST UNIT

Fig. 1 shows a schematic diagram of the unit and a portion of its instrumentation. The A.S.H.V.E. Research Laboratory engineering staff made certain changes during the course of this study and completed most of the instrumentation. Fig. 2 shows a view of the unit and test equipment.

Basically, the unit consists of a pressure atomizing type burner firing axially into a cylindrical water jacketed combustion chamber, a portion of which is

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[1] Exponent numerals refer to References.

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

lined with refractory material as shown. The nozzle has a nominal capacity of 1 gph and produces a hollow-cone spray. Flame length was determined by adjusting the movable water-cooled baffle and noting the position at which the

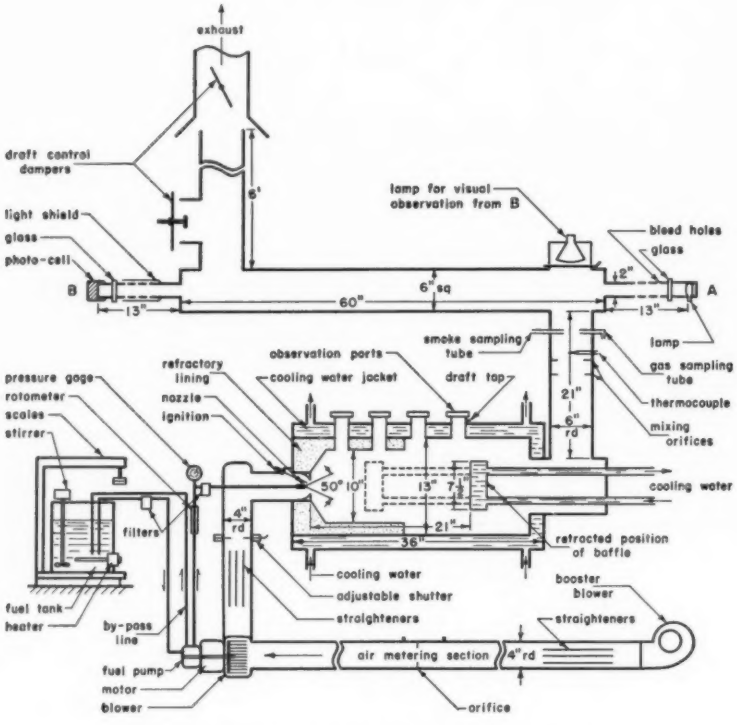

FIG. 1. SCHEMATIC ELEVATION OF TEST UNIT

flame tips just reached the baffle surface. Air supply and combustion chamber draft were controlled by adjusting (1) the cold air inlet damper located in the stack, (2) the butterfly damper in the stack, and (3) the iris type air supply shutter.

TEST PROCEDURE

Oils Tested: To study the performance of the unit, tests were made on six different fuel oils which were known from field experience to have different burning characteristics. The properties of these oils, designated alphabetically, are shown in Table 1; the latest *Commercial Standard* specifications [2] are also given for comparison.

STANDARD TEST CONDITIONS

Since the purpose of the test unit was to rate oils with regard to their burning characteristics, it was necessary that all oils be tested under identical conditions. It was found impossible, however, to meet this requirement with respect to constant weight rate of oil flow through the nozzle without changing the oil line pressure. An effort to compensate for differences due to density and viscosity by supplying the oil at different temperatures was unsuccessful. Hence the following test conditions, hereafter referred to as *standard conditions*, should not be regarded as indicating identical combustion conditions:

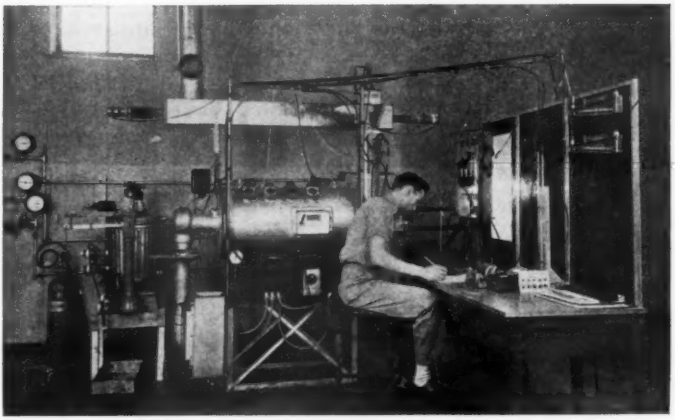

FIG. 2. THE TEST UNIT DURING TESTS

Oil delivery pressure:
100 psig ± 1 psig

Oil supply tank temperature:
110 F ± 2 deg

Cooling water discharge temperature:
145 F ± 5 deg

Combustion chamber draft:
0.020 in. water ± 0.002 in.

Air metering orifice:
2 in. diameter

Baffle position: Fully retracted, except when measuring flame length

Ignition: On

The indicated temperatures were chosen primarily because good control was obtainable at these levels. A warm-up time of approximately one hour was allowed before data were taken, since preliminary investigation showed that this time was required to reach equilibrium.

The performance of each of the six oils was first investigated under standard conditions of operation by at least a dozen tests in which the air supply was varied in steps over the range of about 8 to 15 percent CO_2. These tests are designated as original tests. Check tests over the same range were later made to investigate reproducibility. Tests were then made to determine the effect on fuel rating of different conditions of ignition, turbulence, and oil supply pressure.

TEST RESULTS

In the absence of any primary standard of combustion, the value of the test unit must be judged by its ability to give reproducible results in the measurement of the combustion characteristics of fuel oils, the performance of which is already known in a qualitative way. Measured combustion characteristics are discussed first. In this regard it should be noted that, except for starting and shutdown tests, each value of smoke spot reflectance plotted in this paper is an average of reflectance measurements of four separate filter smoke spots. Each

TABLE 1—PROPERTIES OF THE FUEL OILS INVESTIGATED

FUEL OIL	A	B	C	D	E	F	COMMERCIAL STANDARDS[b]			
							No. 1		No. 2	
							Max	Min	Max	Min
A.P.I. Gravity	39.4	27.5	22.6	43.4	34.0	25.9	—	35	—	26
Flash Point, deg F	146	200	174	134	176	176	—	100	—	100
Viscosity at 100 F										
S.U. sec.	33.0	34.1	33.1[a]	36.5	35.0	35.0	33.3[a]	—	40	—
Centistokes	—	—	2.15	—	—	—	2.2	1.4	4.3[a]	—
Distillation, deg F										
I.B.P.	340	474	378	350	368	418	—	—	—	—
10%	395	511	416	374	433	454	420	—	440	—
50%	446	548	480	425[a]	500[a]	500	—	—	—	—
90%	560	609	548	474	582	567	—	—	675	—
E.P.	616	662	610	518	630	640	625	—	—	—
Pour Point, deg F	-20	—	-45	0	-5	-20	0	—	20	—
Sulfur; %	0.588	0.88	0.275	0.11	0.25	0.36	0.5	—	1.0	—
Water and Sediment, %	Nil	0.1	Trace	Nil	Trace	—	Trace	—	0.1	—
Carbon Residue, % at 10% bottoms	0.036	0.16	0.44	0.207	0.092	0.004	0.15	—	0.35	—
REFINING PROCESS	STRAIGHT RUN	CAT CRACKED	THERM CRACKED	UNKNOWN BLENDS			NOT SPECIFIED			

[a] Estimated values
[b] Specifications from *Commercial Standard* CS 12-48: Ref. 2.

corresponds to a CO_2 determination by an Orsat apparatus made on two successive gas samples. If the CO_2 determinations failed to check, the test was repeated.

Smoke data are plotted against CO_2 for oils A and B in Figs. 3 and 4, respectively; the solid curves are representative of data obtained under the standard conditions of operation. The ordinate scales in these and subsequent figures have been plotted to indicate increasing smoke in a vertical direction. Each of the other oils yielded similar curves. Except for oils B and E there was no marked accumulation of soot in the combustion chamber. In the case of these two oils, a heavy deposit of feathery soot on the refractory occurred above approximately 12.5 percent CO_2 for oil B and about 13.5 percent CO_2 for oil E. An increase in smoke was exhibited by oil B at the higher air supply rates as illustrated in Fig. 4 by the probable trend of the curve for standard conditions.

The results shown by the solid line curves of Figs. 3 and 4 have been replotted in Fig. 5 to summarize the relative smoking tendencies of the six oils tested. All curves except 1 and 2 are for the standard conditions.

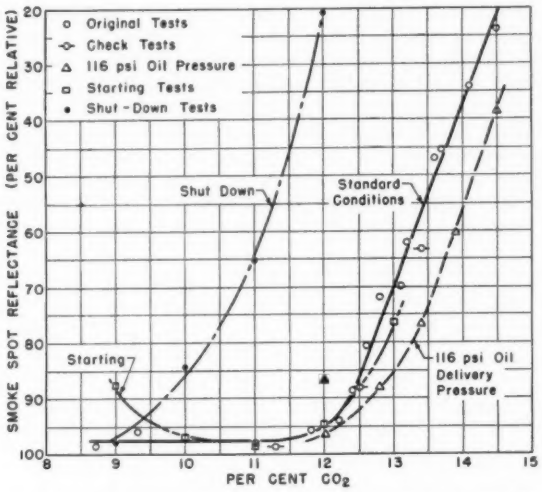

FIG. 3. SMOKING CHARACTERISTICS OF OIL A (15.50 PERCENT ULTIMATE CO_2)

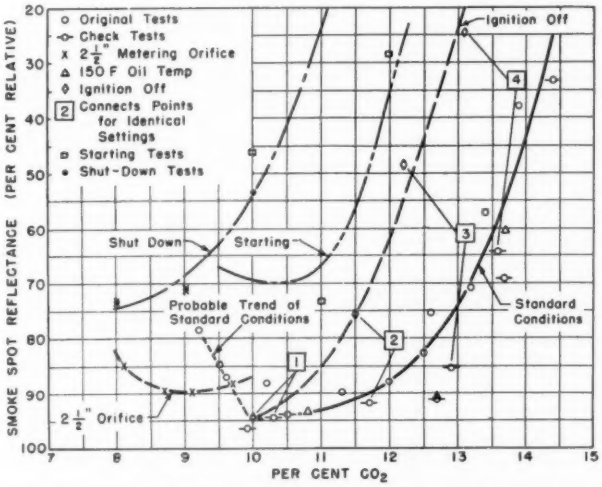

FIG. 4. SMOKING CHARACTERISTICS OF OIL B (16.05 PERCENT ULTIMATE CO_2)

CO_2 in Percent of Ultimate CO_2: Fig. 5 differs from Figs. 3 and 4 in that the abscissa is CO_2 expressed as a percentage of the ultimate CO_2. The ultimate CO_2 for each oil was derived from gas analyses by plotting values of percent O_2 *vs.* CO_2 and extrapolating this straight line to zero O_2. Hydrogen-carbon ratios were determined by their relation to the ultimate CO_2.

By considering smoke density with respect to CO_2 expressed as percent of ultimate CO_2, oils having different hydrogen-carbon ratios can be compared in relation to their actual air requirements. The relation of this concept to the familiar scale of percent excess air also is shown in Fig. 5 for convenience in comparison. Although this relationship theoretically depends to a small extent

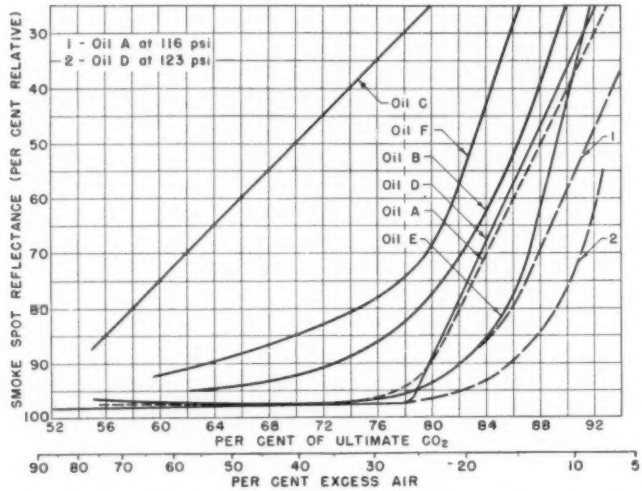

FIG. 5. SMOKE DENSITY *vs.* PERCENT OF ULTIMATE CO_2 AND PERCENT EXCESS AIR

upon the hydrogen-carbon ratio, the range of this ratio for the oils investigated has negligible effect. Use of the concept of CO_2 in percent of ultimate CO_2 has the advantages of spreading the scale in the lower range of excess air, resulting in clearer presentation, and of comparing oils relative to their performance with perfect combustion.

Increased Delivery Pressure: Delivery rates of oils B, C, E, and F when fired at the standard conditions of operation varied less than 5 percent on a weight basis, although slight variations were observed with respect to time and air supply rate. The respective delivery rates for oils A and D at standard conditions were about 6 and 10 percent below the average for the other oils. In order to investigate the relative performance of all oils at similar firing rates, some tests were made with oils A and D fired at higher delivery pressures.

It was possible to increase the firing rates of oils A and D to the level of the other oils by increasing the pressure from 100 to 116 and 123 psig, respectively.

Data obtained with these delivery pressures, other conditions remaining standard, are plotted in Fig. 3 for oil A and are shown as dashed curves in Fig. 5 for both oils A and D; it may be seen that a material decrease in smoke production accompanied the increase in delivery pressure. Stack temperatures were increased about 40 F by the higher delivery pressures. (Indicated stack temperatures for tests at standard conditions varied from 850 to 950 F, depending upon the excess air supplied.)

For the different oils investigated it was found that the effects of delivery temperatures and pressure on nozzle delivery rate could be approximated by

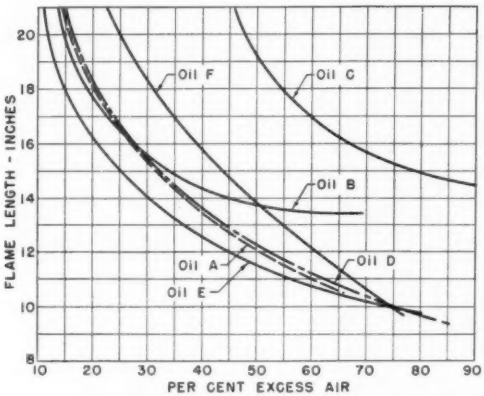

FIG. 6. FLAME LENGTH SUMMARY (STANDARD CONDITIONS)

considering the weight rate of oil flow to be proportional to the square root of the product of density and delivery pressure.

Flame Length: The flame lengths for the various oils were closely related to their smoking tendencies. Fig. 6 shows summary curves of flame length *vs.* percent excess air for the standard conditions of operation. The flame length at constant values of excess air and the smoke corresponding to a constant flame length are shown in Table 2, tabulated in order of decreasing *A.P.I.*† gravity for the six oils.

When oil D was fired at 123 psig the relationship of flame length and excess air became practically identical to that of oil E fired at 100 psig; no appreciable change in flame length characteristic was obtained when the firing pressure of oil A was increased from 100 to 116 psig. The flame length data of the various oils should not be considered as wholly indicative of their relative burning qualities since variable characteristics of ignition, atomization, and mixing may affect the flame length of one oil more than another.

Effect of Air Shutter Opening: For metering the higher rates of air supply a 2½ in. diameter orifice was used, making it necessary to set the air control

† *American Petroleum Institute.*

shutter at a smaller opening in order to keep the air supply system resistance the same. It became evident that the relationship between smoke spot reflectance and percent CO_2 was altered when a larger metering orifice was used; this is illustrated in Fig. 4.

The effect of air shutter opening seemed so significant in the operation of the unit that it was investigated further by several tests using oil B at a constant air-fuel weight ratio of 20.9 and a draft of 0.10 in. water. By dampering the inlet to the booster blower the shutter could be set at different positions without changing the air flow rate or combustion chamber draft. Results of these tests, plotted in Fig. 7, indicate that there was a marked effect of shutter opening upon combustion.

It was believed that the air turbulence set up by different shutter openings persisted into the combustion chamber and was responsible for changing combustion by altering the characteristics of air-fuel mixing.

TABLE 2—FLAME LENGTH OF VARIOUS FUEL OILS

OIL DESIGNATION	D	A	E	B	F	C
A.P.I. Gravity—deg	43.4	39.4	34.0	27.5	25.9	22.6
Flame Length (inches)						
For 25% Excess Air	16.6 15.0[a]	16.6 16.6[a]	15.0	16.5	20.0	>21.0
For 50% Excess Air	12.2 11.5[a]	12.0 12.0[a]	11.5	13.8	13.8	19.2
Smoke Spot Relative Reflectance At 16-Inch Flame Length	98 96[a]	95 96[a]	91	83	84	79

[a] Higher than standard delivery pressures.

Ignition off: Due to occasional fouling of the ignition electrode tips, arcing took place across the electrodes away from the tips and in certain cases arcing ceased completely. In an attempt to determine the maximum effect of this faulty ignition several tests were made with oil B, first with the ignition on and then with the ignition off. These data are plotted in Fig. 4; the numbered lines connect data obtained with ignition on and with ignition off at a given setting of controls. It was noted that there was a marked increase in air supply rate when the ignition was turned off, and this resulted in a decreased CO_2, although smoke density was considerably greater.

Therefore it appeared that the method of maintaining ignition, whether by electric arc combined with radiation and flame propagation or merely sustained by radiation and flame propagation, has a definite bearing upon air flow to the combustion chamber and upon smoke production.

Starting and Shutdown: During the investigation, smoke densities occurring at starting and shutdown were measured in order to indicate the relative smoking tendencies of the various oils at these times. The capacity of the fuel line, increased by instrumentation and filters, may have contributed to sluggish starting and oil cut-off to such an extent that smoking was greater at these times than normally exists in practice.

When starting the unit from cold, the smoke sampling unit was started simultaneously with the pump and fan motor; a second sample was then taken immedi-

ately after the first as a comparison of starting to steady operation after starting. Similarly, prior to shutdown a smoke spot was taken and then the sampler was started as the unit's motor was switched off. Smoke spot reflectance data for two oils at starting and shutdown are plotted against the steady operating values of CO_2 in Figs. 3 and 4. (A few points which aided in establishing the curves fell beyond the range shown.) Smoke density during the starting period was greater than during steady operation, with the exception of oils A and C, but

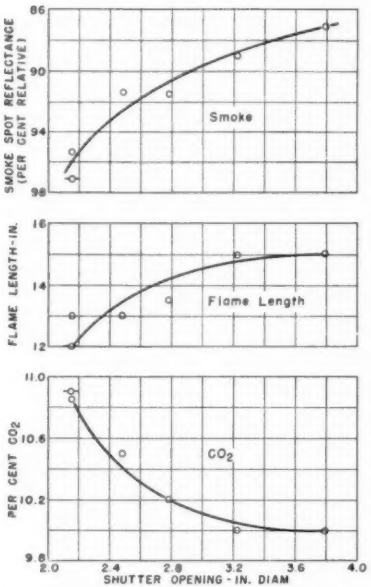

FIG. 7. EFFECT OF SHUTTER OPENING AT CONSTANT AIR-FUEL RATIO

was less than during shutdown. Other data show similar results for conventional burners.[3,4] Impingement or dribble of the oil upon the refractory surfaces occasionally was observed at shutdown.

Reproducibility of Results: Satisfactory reproducibility was obtained for all of the oils except oil C, for which the scatter of smoke data for both original and check tests was as much as 30 percent relative reflectance. Oils C and D burned with greater instability than the other oils, in that detonation and pulsation were much more prevalent. When firing oil D at 123 psig pressure this unsteady combustion sometimes became quite severe.

One factor contributing to the lack of reproducibility for some tests is believed to be the variation of ignition characteristics. Spark position could not be

observed while the unit was in operation so the actual condition of ignition during tests was unknown. Gumming of the electrodes could be observed only after shutdown. It is conceivable that variations in spark position could cause deviations in smoke density approaching the ignition-off values (previously

TABLE 3—COMBUSTION INDICES AND TEST RESULTS

OIL DESIGNATION	NOTE	D	A	E	B	F	C
COMBUSTION INDICES							
A.P.I. Gravity, deg	a	43.4	39.4	34.0	27.5	25.9	22.6
Aniline Point, deg F	a	147	143	137	121	89	55
Diesel Index	b	63.8	56.5	46.6	33.3	23.0	12.4
I.P. Smoke Point	a	22	21	11	8	6	8
Burning Index	c	99	89	71	39	36	23
Cetane Number	d	45	50	51	39	29	19
U.O.P. Charact. Factor	e	11.83	11.78	11.56	11.37	10.99	10.66
TEST RESULTS							
Ultimate % CO_2, U	f	15.35	15.50	15.60	16.05	16.15	16.40
Hydrogen-carbon Ratio	g	0.1525	0.1474	0.1433	0.1275	0.1244	0.1161
Percent of Ult. CO_2							
For 80% smoke spot refl.							
At standard conditions	h	81.5	82.0	85.5	79.0	74.5	58.0
At similar firing rates	i	89.0	86.0	85.5	79.0	74.5	58.0
For 50% smoke spot refl.							
At standard conditions	h	87.5	88.0	89.0	86.1	83.0	70.0
At similar firing rates	i	93.0	91.5	89.0	86.1	83.0	70.0
Smoke Spot Rel. Reflect.							
For 80% of Ult. CO_2							
At standard conditions	h	88.0	89.5	93.0	77.5	68.5	25.0
At similar firing rates	i	96.5	93.5	93.0	77.5	68.5	25.0
For 87% of Ult. CO_2							
At standard conditions	h	52.0	55.0	72.0	46.0	21.0	6.0
At similar firing rates	i	87.5	75.0	72.0	46.0	21.0	6.0

Notes:
(a) Values taken from oil analysis submitted by refiners.
(b) Diesel Index = (*API* gravity x aniline point) /100.
(c) Obtained from chart (Fig 4 of Ref 5).
(d) Obtained from graphical solution chart (Ref 6) based on Institute of Petroleum formula:
cetane number = 175.4 log (mid pt) + 1.98 *API* deg − 496
(e) U.O.P. Characterization Factor = (Cubic Av Boiling Pt, Deg F + 460)$^{1/3}$/Specific Gravity at 60 F (Refs 7 and 8)
(f) Determined from gas analyses.
(g) Calculated from relation: $H/C = (20.9/U - 1)0.421$.
(h) Data obtained when all oils were fired at 100 psig pressure and 110 F (Fig. 5).
(i) Data obtained when all oils were fired at essentially the same weight rate. (Oils A and D fired at pressures greater than 100 psig) (Fig. 5).

illustrated in tests of fuel B) as a limiting condition; this might account for some of the dispersion of test points.

CORRELATION OF TEST RESULTS WITH COMBUSTION INDICES

The results obtained were compared with certain of the indices which are generally considered to be indicative of the burning qualities of fuel oils. The comparison is shown in Table 3 where the six oils are arranged in order of decreasing *A.P.I.* gravity; values of the indices were obtained as indicated in the notes. The test results given in the tabulation were taken from Fig. 5. For

purposes of this comparison the smoking tendency of each oil is based on CO_2 in percent of ultimate CO_2 corresponding to smoke spot relative reflectances of 80 and 50 percent. Also shown is the smoke spot reflectance for CO_2 values of 80 and 87 percent of ultimate CO_2; these values correspond respectively to excess air quantities of 23.5 and 14.0 percent. Results are compared both on

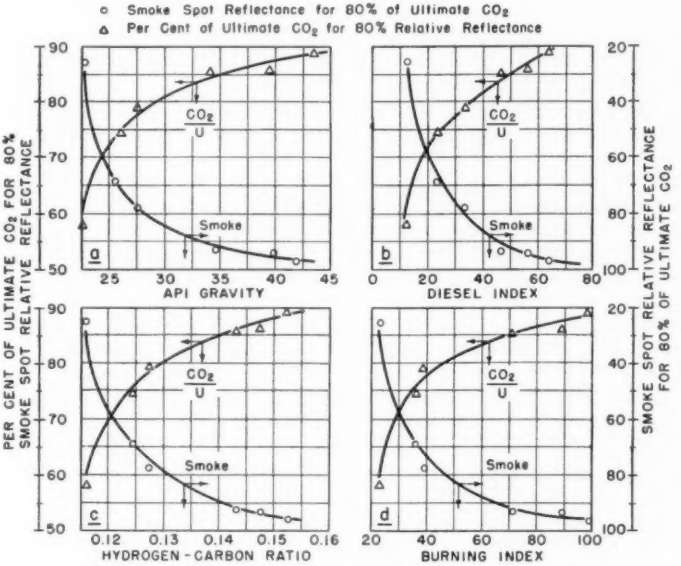

FIG. 8. CORRELATION OF RESULTS WITH VARIOUS COMBUSTION INDICES

the basis of the standard conditions of operation and on the basis of similar firing rates obtained when the oils A and D were fired at pressures greater than 100 psig.

The curves of Fig. 8 were prepared to permit greater ease in visualizing the correlation of the test results with the various combustion indices. Results for similar firing rates are plotted in Fig. 8 against the following indices:

1. *A.P.I.* Gravity
2. Diesel Index
3. Hydrogen-Carbon Ratio, H/C
4. Burning Index

Values of CO_2/U (where U = ultimate CO_2) plotted in Fig. 8 are the maximum values of percent of ultimate CO_2 which may be attained without exceeding smoke densities of 80 percent relative reflectance. (This corresponded

to smoke number 3 on the Bacharach-Shell scale[1].) Also shown are smoke densities obtained at $CO_2 = 80$ percent of ultimate CO_2.

A.P.I. gravity, hydrogen-carbon ratio, diesel index, and burning index seem to be the most indicative of the burning qualities of the oils tested, while U.O.P.* characterization factor and cetane number show a weaker correlation. It may be observed from Table 3 that very poor correlation was obtained with the I.P.†† Smoke Point. The aniline point became more indicative of the burning quality when combined with the *A.P.I.* gravity in the form of the diesel index. Although technique and conditions of operation in the preliminary investigations differed somewhat from those described here, the results with those oils followed the correlations of Fig. 8 quite closely.

Several points of caution should be emphasized in making generalizations from these results and correlations. Variables which affect combustion are so inter-related that it is impossible to control each independently. For example, the comparison of data on the basis of similar firing rates has the disadvantage of including effects of unequal delivery pressures on atomization and mixing. When the oils are considered on the basis of the standard conditions of operation (constant delivery pressure) the relative smoking tendencies are seen to be different from those at similar firing rates and thus correlate differently with the burning indices shown. The air shutter position required for a given percent of excess air varied somewhat with different oils, tending to change the relative air-fuel mixing characteristics (Fig. 7). Due to differences in oil spray characteristics, the optimum requirements of ignition position could possibly vary for the different oils. These factors tend to indicate that the data presented should not be considered to be comparable in the strictest sense.

The data presented are the results obtained for the *oils used in this investigation* and do not necessarily indicate the relative intrinsic burning qualities of these fuels, nor does the correlation with the combustion indices indicate that such a correlation could be applied for all types of oils. It should be pointed out that relatively few oils have been investigated and that the burning qualities of a number of widely different oils of the same type, such as catalytically-cracked oils, might not follow the correlations presented. These correlations are encouraging however, and it is possible that they could be firmly established by tests with a greater number of oils; thus a simple determination such as *A.P.I.* gravity might be found to be indicative of the burning qualities as evaluated by the test unit.

CONCLUSIONS

1. Satisfactory reproducibility of results was obtained with all oils except oil C, for which combustion was quite unstable.

2. In addition to increasing the firing rate, the effect of increasing the delivery pressure of the oil was to produce cleaner combustion and a slightly shorter flame.

3. At constant air-fuel ratio the effect of decreasing the air shutter opening was to reduce smoke density and flame length, while increasing percent CO_2.

4. The effect of switching the ignition off during steady operation was to increase the air supply rate and to increase smoke production.

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* Universal Oil Products Co.

†† International Petroleum Technologists.

5. Smoke densities during shutdown were greater than during starting, and smoke densities during starting were generally higher than during steady operation.

6. A good degree of correlation was obtained between test results at similar firing rates and the following combustion indices: *A.P.I.* gravity, hydrogen-carbon ratio, diesel index and burning index. A greater number of oils must be investigated if these correlations are to be so established that a simply obtained index can be used to rate the burning qualities of fuel oils. Investigation should be made to determine whether these correlations may be extended to other types of burners and to field installations.

7. In order to determine the significance of the results obtained with the test unit, it would be desirable to investigate fundamental principles of fuel atomization, ignition, air-fuel mixing, and smoke measurement.

ACKNOWLEDGMENT

The authors acknowledge with thanks the cooperation and suggestions received from their colleagues of the A.S.H.V.E. Research Laboratory staff and members of the Engineering Committee of the *Oil-Heat Institute of America, Inc.*

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DISCUSSION

E. F. OPPERMAN, Stamford, Conn., (WRITTEN): This paper and the paper preceding (Chapter No. 1412) are very interesting and appear to propose a reliable and simple method of analyzing the burning characteristics of an untried fuel oil. The research to date has very promising results from the testing of six oils in a test unit incorporating one burner design. This, therefore, would simulate field expectancy if the unit in the field were of the exact design as the test unit. Unfortunately, this will not be the case and with the hundreds of manufacturers in this business the variables are numerous and all will produce widely different burning characteristics.

A combination of one manufacturer's burner using a certain fuel oil may produce excellent results. However, this same oil when used with a burner of different manufacture may produce a very poor flame and in extreme cases may not work at all. That is, various types of burners have entirely different burning characteristics when using the same fuel oil. Some of these different charcteristics are attributed to: (1) different characteristics of nozzles used; (2) type of ignition whether constant or intermittent; (3) type of mixing head; and (4) quality of oil atomization.

These few variables are mutiplied a hundred-fold by conditions of manufacture. For instance, two apparently identical nozzles constructed in accordance with identical specifications, but of different manufacture, have been found to produce very different results.

It seems, therefore, that the method as proposed has definite possibilities, however, variable conditions of manufacture in addition to those variables encountered in the field would indicate that a great deal of additional research will be necessary.

N. A. PALMER, Bloomington, Ill. (WRITTEN): Much valuable data are contained in the report of this project and the authors are to be commended on the excellent job. I am in full agreement with their conclusions that this report points up the need for further research on the subject.

It becomes evident that a number of factors directly affect the amount of smoke production in any given fuel oil, such as: degree of atomization; spray pattern and angle of the nozzle; and pressure and turbulence of the air.

Another ever present question arises with respect to the reduction of the smoke by introducing excess air. Does it actually reduce the production of smoke or merely dilute it? If dilution is the result, we would have reason to question the reliability of the smoke density curve in Fig. 5.

It would seem desirable to design or select test equipment to produce no smoke at ultimate CO_2 with a prototype or reference fuel of known burning characteristics. Then, a fuel oil under test could be classed as sub-standard if the introduction of excess air to acceptable limits did not reduce or eliminate smoke production.

A test unit of this nature could be of great value to the oil burner industry in proving the need for improved equipment design practices.

I believe the ultimate objective should be the inclusion of minimum performance requirements in *Commercial Standard* CS75.

T. H. SMOOT, Marshalltown, Ia., (WRITTEN): My principal comment has to do with the conclusion that when the burner ignition is turned off during steady operation, both the air supply and smoke production increased. Although these results did prevail in the test work reported in this paper, the impression should not be given that they are inherent conditions and will apply to all oil burners of the high pressure atomizing type. It is my opinion that these observed conditions took place because the point of flame retention or flame front moved away from the burner with ignition off. The burner used in the test unit does not incorporate design characteristics which make its point of flame retention independent of the ignition means. Burner mixing heads are in use which do control the location of the flame front and with which there is no observed movement of the flame front whether ignition is on or off. It is my opinion that with such burner heads there would not be increased air supply and smoke production when ignition is turned off. This characteristic of flame retention is largely independent of fuel quality.

Smoking tendencies on starting and stopping are principally the result of the mechanical functioning of burner parts rather than having to do with fuel quality. These conditions can be controlled by additional components on the burner so as to provide for delayed fuel delivery at the start and instantaneous fuel shut-off when the burner stops.

The burning qualities of fuels on the basis of short runs are of great importance in view of the increased false heat input to thermostats (*i.e.* heat anticipation) where maximum cycles of 3 min duration are finding widespread application. The burning qualities on the short cycle basis, in addition to the steady run basis, should be evaluated in the fuel burning index.

M. F. DIXON, Toronto, Ont., Canada: I congratulate the authors in trying to set up a scale of fuel evaluation. From my knowledge of the subject, I realize it is a very difficult task.

The first question that comes to my mind, as more work is done, is: What is the next step? Would there actually be specifications indicating a certain smoke density, measured by one of the four methods previously discussed here, or would that be a specification for the fuel oil buyer with his supplier?

Secondly, if you did have such a specification, where is the actual testing going to be made or done? If it is done at the refinery laboratory and you have a certain maximum smoke density, allowable for a certain type of fuel, will that specification stand up at the time the fuel is being used, which may be a year later, because the refiner makes his fuel in the summer and it is probably not used until the following spring?

That brings up questions of potential sediment-forming characteristics of the fuel. One fuel that has a smoke pass in the refinery laboratory might utterly fail to pass it when used in the field.

I can think of so many problems in connection with this, if I did see results of a particular fuel having certain smoke-forming tendencies, I would want to know, for example, was it a field test or laboratory test? What was the potential sediment-forming characteristic of the fuel? Finally, what was the fuel—straight run, mixture of thermally cracked or catalytically cracked gas oil, and were the two components stable or were they not, or did they tend to become unstable later?

L. N. HUNTER, Johnstown, Pa.: This work was not done at the laboratory with the intention or purpose of testing anyone's oil or anyone's oil-burner. The main purpose of the test was to try to evaluate this instrument as a means of measuring the burning qualities of fuel oil. It was hoped that we might develop an instrument for measuring the burning qualities of fuel oil similar to the engine that is used in the gasoline industry to measure the anti-knock qualities of gasoline. The cost of the work was borne by the *Oil-Heat Institute of America, Inc.*

Regarding the future status of this development the *Oil-Heat Institute* has a committee which is reviewing these results, and, after they have made their review, they will make recommendations as to whether further work should be done or just how this project should be disposed of. If further work is recommended, the A.S.H.V.E. Research Laboratory will probably be given an opportunity to continue to do the work, if they so desire.

On behalf of the *Oil-Heat Institute* we want to acknowledge the excellent work which the Laboratory has done on this program so far.

F. R. DUNN, JR.*, Philadelphia, Pa.: We are very much interested in this work and I would like to add my congratulations. We have followed the course of the work on the test unit in the several A.S.H.V.E. reports that were issued to *Oil-Heat Institute* members. In general, the information in this paper ties in rather well with the sort of information that we have obtained in our work on oil burning. Our work has not been done in a special design unit such as the test unit, but, rather, on commercial burners for domestic use. We have developed and used the Atlantic burning index[5] to distinguish between the relative burning quality of fuel oils produced from various

* Research & Development Dept., ATLANTIC REFINING CO.

sources. We have attempted by the burning index to define the burning quality of a fuel oil in terms of the range of operating conditions, air rates and oil rates, over which a typical domestic burner will operate without producing more than barely visible smoke at a stack exit. We have approached the problem from that standpoint simply because in the field the satisfactory operation of burners with clean burning is contingent upon proper burner adjustment and inadvertent changes in air and oil rate caused by fluctuations in stack draft, oil temperature, etc. Obviously, a fuel oil capable of burning cleanly over a wide range of operating conditions is superior in performance to a fuel oil which can burn cleanly only over a small range of operating conditions.

Therefore, we have looked at the problem of determining the burning quality of fuel oils not so much from the standpoint of how an oil will burn under just one set of operating conditions, but, rather, how an oil will burn over a wide range of operating conditions in various types of domestic burning equipment.

There are probably several hundred brands of domestic burners in use today, all of which can be classified into a few types, such as high- and low-pressure burners, rotary wall-flame burners, and pot- or space-heating burners. On the basis of tests which we have run we have been able to set up specifications for the proper fuel oil for each one of these types of burners.

The information gained with the test unit is excellent. It has been demonstrated in the test unit that most types of fuel oils, with the exception of the thermally-cracked oil, give satisfactory performance in the less than 11 percent CO_2 range. From the curves in Fig. 5 it is noted that high smoke values are not obtained until relatively low excess air is used; we have had this same experience in our tests made in domestic burners of a type roughly similar to the test unit.

We have also noted that there is a big difference in certain types of equipment in their ability to handle fuel oils, but, again, many of these differences are encountered at the higher CO_2 values.

F. J. NUNLIST, Milwaukee, Wis.: I would like to add my congratulations to the authors of the paper, but I would like also to add this thought: I think it is clear that some additional work is indicated. These results as published do open up many fields of investigation.

For one, I am curious about the test oil C and its failure to fall into a pattern. I wonder, if there are other oils along that line, whether or not the index is going to have true value for us in terms of interchangeability of fuels.

Secondly, I am concerned about this ignition-on and ignition-off performance, which implies that perhaps there was some change in power characteristics, or perhaps there was a change in the speed of the pump, the oil delivery rate, or something which may or may not have happened but does not seem to be indicated in the paper.

Then I am also concerned with the plotting of the two oils at 116 and 123 psig rate which could, as shown in the earlier curve, give different performance characteristics. It might perhaps be better to have changed the nozzle sizes to maintain the same kinetic energy at the nozzle.

Finally, it seems to me that it would be very desirable in the three cases of the oils of undetermined origin to know something about their origin and what the original composition might have been. Perhaps they did not have any of the thermally-cracked oil which fell off the curve, and if they had, different results might have been obtained. Obviously, if an index can be developed, it is going to be of considerable use to application engineers and likewise to designers of oil-burning equipment.

K. O. BEATTY, JR., Raleigh, N. C.: The work reported in this paper is a fine start in a good direction. In our furnace oil evaluation work at North Carolina State College we have been using four different types of domestic oil burning units, a

rotary wall-flame burner, a natural draft-pot burner, a forced draft-pot burner, and a gun burner.

Our experience with a gun burner has been that it could be adjusted to burn satisfactorily with most any type of light furnace oil. The other types of units are much more sensitive. Although, I recognize the importance of the gun burner because of its wide use, I wonder whether some other more sensitive type of unit might be worth considering.

I note that when you used a lighter fuel you found it necessary to step up the oil pressure. What is your explanation of this?

In Fig. 8 you show four very good correlations between smoke and certain properties of the oils. We have used a very wide variety of oils in our work and have never been successful in finding as good agreements as these with any properties.

AUTHORS' CLOSURE (D. W. Locklin): The comment of Mr. Dixon pertains to the application of the test unit once it has become a proven instrument. As Mr. Hunter has pointed out, the Engineering Committee of the *Oil-Heat Institute* envisioned the use of this unit in a manner similar to the CFR octane engine, whereby burner manufacturers, oil refiners, or anyone concerned with the problem could establish a combustion rating using a basic design of this unit and a prescribed procedure. Such a rating, possibly in terms of a reference oil, might well be used as a contract specification. It is hoped that, by considering this test unit as a primary instrument, a secondary test or index can be established; this would permit evaluation of fuel oils in the field by simpler and speedier means.

The problem of storage stability is believed to be more of a handling problem than one of combustion, although sediment in lines and filters can certainly effect burner operation. The actual concentration of sediment formation normally encountered is probably not sufficient to alter the physical properties or burning quality of the remaining fuel. The increasing use of deeply cracked stocks in domestic fuels has brought about new problems in fuel stability and has, therefore, increased research activity in this field.

The Burning Index (See Reference 5) discussed by Mr. Dunn is compared to results of this investigation in Fig. 8. It is significant that this index of adjustment tolerance, obtained with a rotary wall-flame burner, showed such good correlation with the performance obtained in the test unit.

It is quite possible, as Mr. Nunlist suggests, that other oils might behave as oil C; however, this oil was beyond the specifications for No. 2 grade and far beyond the limits of current fuels for domestic burners.

Conversation with other investigators has shown that they also observed a change in air supply when the ignition was shut off. The possibility that the fan speed was affected by the ignition-off condition was not investigated.

Nozzle performance appeared to play a major part in combustion characteristics of the unit. In order to minimize this variable of atomization, it was deemed desirable to run all tests with a single nozzle. Preliminary tests indicated that atomization varied each time the nozzle was disassembled for cleaning; therefore, all tests reported were run without disassembling the nozzle.

The three oils of undetermined origin were procured locally in order to obtain typical burner fuels for comparison with the straight run, catalytically cracked, and thermally cracked oils supplied by refiners. In general, the composition as to refining method is not believed significant in itself since there may be wide differences in stocks by the same type of process.

Professor Beatty has suggested the possibility that another type of burner might be better suited for evaluation of fuel oil burning quality. Burners which normally handle both No. 1 and No. 2 grades of fuel are the gun type burner and the rotary wall-flame burner. As a laboratory tool for fuel oil evaluation the wall-flame burner

offers certain advantages: it is, in general, more sensitive to fuel oil quality when considered at a given set of conditions, and it permits control of firing rates without introducing other variables of atomization and flame retention.

To determine temperature and pressure effects upon oil rates, an analysis of the firing rate data of this investigation was made on the basis of orifice discharge theory. This showed that the nozzle discharge coefficient, over the range of oils and rates investigated, was independent of Reynolds number and therefore viscosity; for the relatively narrow range of properties represented in these tests, the weight rate was found to be nearly proportional to the square root of the product of firing pressure and oil density. In order to maintain similar firing rates it was thus necessary to raise the firing pressures for the lighter oils.

Dilution effects mentioned by Mr. Palmer are included in Figs. 3, 4, and 5, but such dilution is present over the entire range of CO_2 and does not occur suddenly as suggested by the smoke curves. The change from 10 to 15 percent CO_2 represents a variation in the total weight of flue gas amounting to only about 25 percent; thus, it is believed that dilution effects together with variations in smoke sampling efficiency were not sufficient to mask the relative smoking tendencies of these oils.

The data at constant air-fuel ratio shown in Fig. 7 permit the estimation of amounts of unburned carbon in the flue gas represented by various smoke readings. If the oil is considered a pure hydrocarbon burning without CO, the reduction of CO_2 implies an increase of free carbon amounting to about 7½ percent by weight of the fuel fired or about 5 perecnt of the potential heating value of the fuel. This analysis offers an indication that the concentration of free carbon represented by an increase from Smoke No. 1 to No. 2 on the Bacharach-Shell scale, increased about 0.004 lb per lb of flue gas.

This investigation succeeded in proving that the test unit and the technique developed was capable of distinguishing between the combustion performance of the oils studied. The question prompted by Mr. Opperman is then whether such an indication of relative performance reflects the inherent burning quality of the oil itself—and its performance when fired in other burners under different conditions. Different burners firing the same oil will, of course, give different conditions of absolute performance but the significant question is whether the unit shows the same *relative* performance as do other burners. If this can be established by further laboratory tests and field experience, the value of the unit will be proved.

The authors are in agreement with Mr. Smoot's conclusion that the change in combustion air supply, reported with ignition *off*, was related to the position of the flame front and that such a variation in air rate would probably not occur with burners having flame retention independent of ignition. It is also believed that stable flame retention is important for the elimination of pulsation.

Mr. Smoot has pointed out that the method of evaluation by the test unit was based upon performance at *equilibrium conditions*; starting and shut down conditions were not fully studied. From the standpoint of sooting during normal operation of a burner the critical periods are the transient conditions at the beginning and end of a burner cycle, unless special corrective measures are taken. Thus, in order to obtain a true picture of fuel performance under field conditions, investigation will have to be extended to the evaluation of fuels burned in intermittently operating burners.

1414

No. 1414

A SURVEY OF ELECTROSTATIC PRECIPITATION

By ERIC A. WALKER* AND J. E. COOLIDGE,** STATE COLLEGE, PA.

ALTHOUGH one can find references to electrostatic precipitation and attraction of charged dust particles by Gilbert, as far back as the year 1600, the first comprehensive outline of the problem was presented by Lodge in 1884. Lodge claimed industrial dust in the atmosphere was becoming an increasing hazard and demonstrated its removal by means of an electric spark. His attempt at practical application however was not a success; in the most part because of an unfortunate choice of material: choosing lead fume which is now known to be among the most difficult solids to precipitate.

In 1911 Cottrell constructed the first successful electrostatic precipitator. His success was aided by new high voltage techniques developed since Lodge's attempt and a more fortuitous choice of material, sulfuric acid mist, to precipitate.

From this slow start electrostatic precipitation has grown to an immense business in both this country and Europe and many major installations have been made to precipitate a wide variety of materials. The smallest are household ventilation units while the major units include such giants as fly ash installations handling 200,000 cfm and precipitating some 200 tons of ash per day.

Unfortunately, an understanding of the fundamentals involved in the process has not had such an impressive growth. Some effort is being directed toward remedying this situation but electrostatic precipitation must still be classed as an art, not a science.

METHOD OF OPERATION

The simplest type of treatment chamber consists of a wire and concentric cylinder with a high voltage maintained between the wire and cylinder to produce a voltage discharge. The dust laden air is passed through the cylinder where the dust is charged by the corona discharge, and is then transported by the associated electric field to the cylinder where it can be collected. The dust is removed by mechanical rapping or flushing with an appropriate fluid. A

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variation of this arrangement is a wire flat-plate unit. The principle of operation is the same.

Mechanical commutator power supplies are widely used because of the high voltage they will rectify (as high as 100 kv) with few failures. Such units have the disadvantages of radio interference, high maintenance costs and fire hazard to the extent that vacuum tube type rectifiers (used successfully in lower voltage applications) are rapidly replacing them.

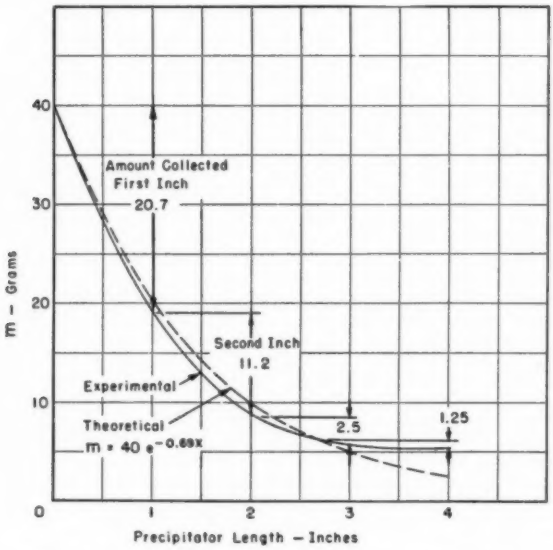


FIG. 1. PRECIPITATOR COLLECTION *vs.* PRECIPITATOR LENGTH

THEORIES OF ELECTROSTATIC PRECIPITATION

In many applications it is advisable to treat precipitation as a single operation and express its action in terms of its overall physical factors, *i.e.*, the amount of dust put into the precipitator, the length of time it is under the influence of the precipitator, etc. Under these considerations an equation for the efficiency of precipitation can be written as

$$n = 1 - e^{-x} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where

n = the ratio of dust removed to the dust which enters
x = a factor involving many variables

Such an expression has been derived by Anderson, Horne, Deutsch and possibly others. It is of limited use especially in predicting the behavior of completely new materials and conditions. The general form of the expression appears in

accord with the results shown in Fig. 1 in which x becomes a distance along the precipitator. This curve was obtained with the experimental precipitator shown in Fig. 2.

A consideration receiving increasing attention to make units more efficient is gas distribution. It is obviously desirable to make each section of a precipitator bear equal load.

An understanding of the individual processes of electrostatic precipitation is essential to intelligent design and improvement in operation. The following discussion is a brief outline of these processes along with their status.

Precipitator Corona: In view of the great amount of work which has been devoted to the study of corona, surprisingly little has been done on corona in

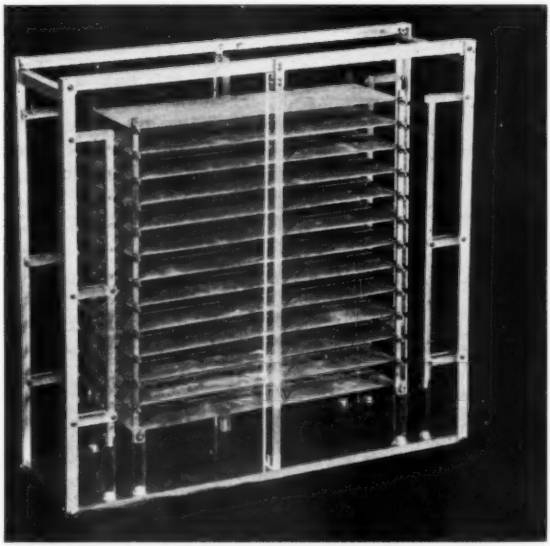


FIG. 2. EXPERIMENTAL PRECIPITATOR

a gas contaminated by fume or dust. This contamination alters the discharge tremendously as will become evident.

Conditions are quite different in negative wire corona, from those in positive wire corona. In general the differences can be given as: (1) negative wire corona is highly localized, depends on the characteristic of the wire for a sustained discharge, is bead-like in appearance, produces ozone; (2) positive wire corona is a broad discharge, depends on the gas—not the wire—for being sustained, has a glow-like appearance, reduced ozone; (3) for clean metal electrodes, the arc-over voltage for negative corona is about twice that of positive.

The mechanisms of positive wire corona are susceptible to theoretical analysis; the current and electric field can be calculated from the geometry of the arrangement and the applied voltage. However, the capricious nature of the sustaining

mechanism of negative wire corona allows no simple general formula to be derived either for the current or the electric field.

The presence of the dust in a precipitator makes additional considerations necessary above that of corona discharge with clean electrodes. The previous history of the precipitator, the gas velocity, the humidity, the composition of the gas and the other factors which are less comprehensible alter the electric discharge. Fig. 3 represents the dependence of precipitator current on relative humidity before, during, and after a precipitator run with gypsum. The theory

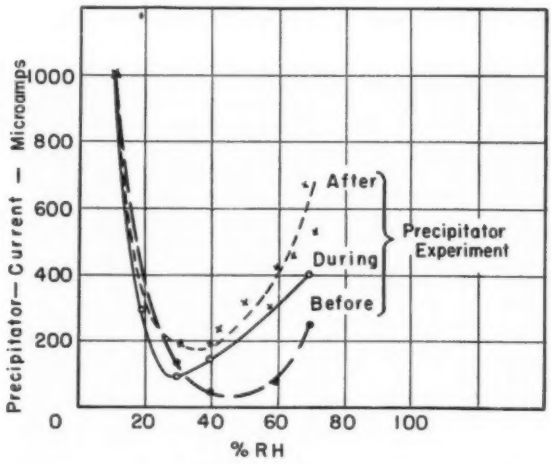


FIG. 3. PRECIPITATION CURRENT *vs.* RELATIVE HUMIDITY
(7750 volts; 3.3 fps; 3 grains/cu ft; 78 F; gypsum)

behind these changes is not understood but they illustrate the pronounced effect of humidity and dust on the electric discharge.

The first important consideration in the design of an electrostatic precipitator is the capacity of the power pack. This is calculated from the length of the corona wire employed. Thus an important design constant is the *milliamperes per foot* at the operating voltage. From our previous discussion it can be seen that this relation is extremely difficult to obtain from theoretical considerations. In addition, it is found by actual measurements that the variation in successive current readings may be as much as 20 percent from reading to reading. If one attempts to measure the current as a function of the length of the wire, results are even more uncertain. Thus power pack design is a case where intuition is an important factor.

One might question the desirability of employing negative corona in precipitators. However, from an operational standpoint its one advantage of higher arc-over voltage outweighs its disadvantages.

The electric field strength distribution in simple geometric shapes in the presence of an electric discharge can be predicted from theoretical consider-

ations for positive wire operation but not for negative. Probe measurements of the electric field are unsatisfactory. An approximate expression for the field strength is:

$$E = \frac{V - V_c}{R - \alpha} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where

E = field strength
V = applied voltage
V_c = critical corona voltage
R = radius of cylinder
α = radius of wire

Application of Charge to Dust: There are two schools of thought concerning mechanism by which the dust particles become charged: (1) the electric field is the main cause of the action; (2) the kinetic energy of heat motion produces the action. Table 1 gives a summary of the theories proposed and their individual considerations. Ladenburg is the principal proponent of the first theory. He states that when the electric charges which are propelled to the particle by the applied field are numerous enough to set up a local field equal to the applied field at the surface of the particle, the charge has reached the maximum. This maximum is given by:

$$n_m = \frac{KE_A r^2}{e} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where

E_A = applied field
r = particle radius
e = electron charge
$K = 1 + 2[(\varepsilon-1)/(\varepsilon-2)]$
ε = dielectric constant

Ladenburg says nothing about the time required for deposit. Pauthenier started from Ladenburg's basic assumption, but by defining an entry zone in terms of the charge on the particle was able to include the variation of charge with time.

Arendt and Kallman considered the kinetic energy viewpoint and were also able to arrive at a value for maximum charge. Deutsch expanded on this idea and considered the charging by kinetic energy and transportation simultaneously.

It is unfortunate that a quantitative discussion cannot be made of a comparison or application of the theories outlined here. In general the electric field proponents arrive at values higher than those from kinetic energy. Although many experiments have appeared to confirm Pauthenier's expression, the method of test and the inability of field calculation place such results in a doubtful light.

TRANSPORT PROBLEM

The calculation of the time necessary to transport a charged particle can be made from a knowledge of the magnitude of the charge, the strength of the electric field, the size of the particle and Stokes' Law for viscous motion. As has been pointed out neither the charge nor the electric field is capable of accurate theoretical prediction at the present time so a transportation calculation is quite difficult. However estimates of transport time, based on existing theory, compared to precipitator exposure time can indicate the possibility of successful precipitation.

TABLE 1—COMPARISONS OF PARTICLE CHARGING THEORIES IN A CORONA DISCHARGE

AUTHOR	STARTING POINT	MAXIMUM CHARGE	TIME TO CHARGE	TRANSPORT TIME
Ladenburg	Copious supply of charges. $E_A = E_p$ at surface of particle. Particles > 1u	$n_m = \frac{E_A r^2 K}{e}$	———	Stokes' Law
Pauthenier	Same as for Ladenburg plus image forces. Defines a permissible particle entry zone for charging. Particles > 1u.	$n_m = \frac{E_A r^2 K}{e}$	$t = \frac{q}{\pi K_v f e\,(K E_A r^2 - q)}$	Stokes' Law. Uniform field superior to exponential for transportation $t_e = \frac{t_o}{2} ln \left(\frac{R_o}{r_o}\right)$
Arendt and Kallman	Kinetic energy required to deposit in field E_A against local field E_p. Particles < 1u.	$e\frac{dn}{dt}\left(1+\frac{xcr^2}{nK_v e}\right) = 4\pi r^2 xcfe^{-\frac{ne^2}{rK_B T}}$ n_m when $\frac{dn}{dt} \rightarrow 0$ - - - - - - - - - - - - when $t = 0$ $\frac{dn}{dt} = 4\pi r^2 xcf$	For small percentage of charge minimum time can be calculated from $\frac{dn}{dt}$ at $t = 0$ $n = 4\pi r^2 xcft$ for $n << n_m$	
Deutsch	Collisions by kinetic energy only. Does not consider E_A.	$ne = \frac{Lr}{e\alpha} ln \frac{ur^2}{3\sqrt{\pi} We}$	———	$W = \frac{L}{6\pi ne\alpha} ln \frac{ur^2}{3\sqrt{\pi} We}$ (Stokes' Law)
Coolidge	Same as for Kallman	$\frac{n_m e^2}{r} \geq L_r$ $\frac{dn}{dt} \rightarrow 0$	$\int_0^n \frac{dn}{\frac{2\sqrt{\pi} r^2 E_A K_v f}{(KT)^{\frac{3}{2}}} \int_{\frac{ne^2}{r}}^{\infty} E^{\frac{1}{2}} e^{-\frac{E}{K_B T}} dE} = \int_0^t dt$	

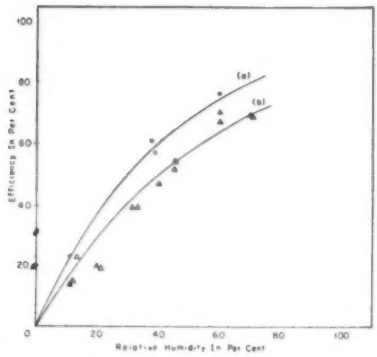
NOMENCLATURE FOR TABLE 1

n = number of elementary charges at time "t"
n_m = maximum number of elementary charges
r = particle radius
$K = 1 + 2\left(\frac{\varepsilon - 1}{\varepsilon + 2}\right)$
e = elementary electron charge
$\boldsymbol{E}_A$ = applied electric field
$\boldsymbol{E}_p$ = electric field of charged particle
ε = dielectric constant
q = charge on particle at time "t" = ne
f = ion density
K_v = ion mobility
c = ion velocity
x = proportion of particles sticking to particle (decimal)
K_B = Boltzmann's constant
T = temperature, absolute
L = mean kinetic energy of charging ions
u = mean molecular velocity
W = particle mobility
$\alpha = \left(1 - \frac{1}{2\sqrt{n}}\right)\left(1 - \frac{1}{\sqrt{n}}\right)$
L_o = maximum kinetic energy of charging ions (arbitrarily chosen)
E = kinetic energy of a charging ion
t = time
t_e = transport time in an exponential field
t_o = transportation time in a uniform field
R_o = collector cylinder radius
r_o = discharge wire radius
ln = logarithm

Removal of Charge from Dust: If the material comes in contact with the collector plate, it is then necessary that the particle give up its charge and assume the charge of the collector plate. If the resistivity of the deposited layer is extremely high this charge transfer becomes difficult, back voltages are built up, sparking and uneven current flow result and precipitation is difficult. Some studies have correlated the resistivity of the precipitated material with the efficiency of precipitation of the material. These studies indicate that if the resistance is less than 8×10^8 ohm-cm precipitation is easy, but if it reaches 10^{13} ohm-cm precipitation is almost impossible.

Many investigators have mentioned the desirability of humidifying the gas in order to improve precipitation under certain conditions. In accordance with this practice laboratory investigations show that relative humidity plays a prominent role in the precipitation of gypsum compound. Typical results are shown in Fig. 4. The experimental precipitator used in these investigations is shown in Fig. 2 and employs thirteen 16-gage, 4-in. × 12-in. plates, spaced 1 in. apart with four wires, 20 mils in diameter, spaced equal distance between each pair of plates. The data suggest a relation for efficiency.

$$E = 1 - e^{-kh(V - V_c)} \quad \ldots\ldots\ldots\ldots \quad (4)$$



(Experimental precipitator; Gypsum No. 1; 3 grains/cu ft; 76 F — 80 F)
(a) 8400 volts — 3.3 fps.
(b) 7750 volts — 3.3 fps.

FIG. 4. EFFICIENCY *vs.* RELATIVE HUMIDITY

where

V = applied voltage
V_c = critical corona voltage
h = fractional humidity
k = constant for the system

This relation is applicable to the gypsum compound only. It appears however to agree with the aforementioned resistivity studies—the effect of the moisture probably giving a reduction in resistivity.

REMOVAL OF PRECIPITATED MATTER

The removal of liquid precipitates is usually not a problem. Solids adhere to the collector plate and must be removed by scraping or shaking. These methods result in re-entrainment of collected material which obviously is undesirable. Much effort has been devoted to preventing such re-entrainment but to the present time a totally effective method has not been devised. It is not the purpose of this article to discuss this particular aspect of the problem since in general it is an engineering one only.

UNSOLVED PROBLEMS

Early in this work it was noticed that dust could be precipitated by energized parallel plates without a corona discharge. Experiments revealed that the action was caused by the frictional electrification of the dust cloud during its dispersion and flow through the entrance duct to the precipitator. Because of the simplicity and low energy requirement of such a system studies were initiated on the properties of frictional electricity. These studies have not resulted in a complete understanding of the phenomenon. However, it is believed that further study will develop the possible applications of frictional charging.

In the course of the studies of frictional electrification with parallel plates a discrepancy appeared between the measured current to the plates and that expected from the amount of material deposited and an estimate of the frictional charge. Further investigation indicated the explanation was electric re-entrainment. Electric re-entrainment can be described as follows: following initial

deposition, the particle assumes the same polarity as the plate. As a result there is a repelling force set up between the plate and the particle because of their like charges. If this repelling force is greater than the mechanical forces holding the particle to the plate the particle will be re-entrained in the air stream. In parallel plate units it will travel to the opposite plate where the same action takes place. Thus a system of bouncing particles exists between the plates. These particles contribute to the current and explain the large magnitude of the measurements. Fig. 5 illustrates the effect of this action where at a field strength of 5000 volts per centimeter, although the rate of particle removal has reached a maximum, the current increases sharply. In this region the electric repelling forces dominate the mechanical adhering forces to a great extent.

The statement is often made that electrostatic precipitators are more adaptable to removing small particles than large ones. Theoretical consideration of particle

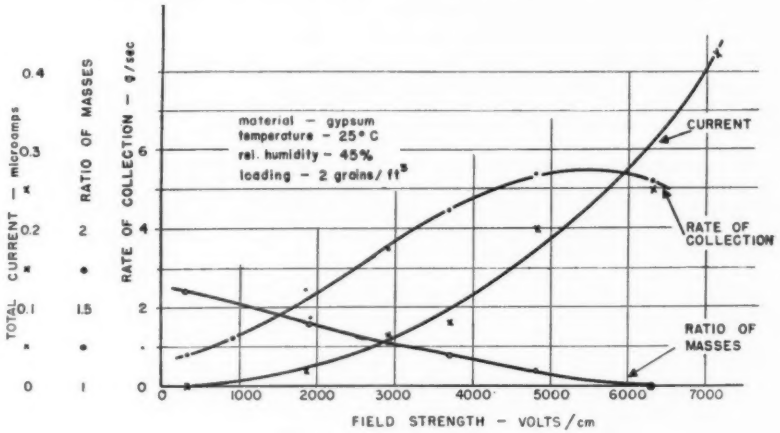


FIG. 5. CHARACTERISTICS OF PARALLEL PLATE PRECIPITATION

charging and particle transportation indicates that the opposite is true. In fact, measurements of the particle size distribution of collected material along the experimental precipitator of Fig. 2 show that the larger particles are precipitated first. Of course, particle size distribution measurements are such that their results bear a somewhat tenuous relation to the actual distribution but measurements conducted in the same manner can indicate relative size.

The Effects of Resistance: It was pointed out earlier in this paper that studies have been made of the effects of the resistivity of the deposit on precipitation of various materials. In the early development of precipitation theoretical analysis was more or less confined to corona discharge, particle charging and particle transportation, but it appears increasingly evident that the action at the collector plate is the limiting factor in many installations. The exact nature of this action is not known and much further study is necessary for adequate understanding.

ACKNOWLEDGMENT

This research program was done under the sponsorship of the Koppers Co. to whom the authors express their appreciation.

DISCUSSION

G. W. PENNEY, Pittsburgh, Pa.: You will note that the curve in Fig. 3 is a curve between precipitator current *vs.* relative humidity. We know that you can get this sort of curve either with the low-voltage type of precipitation used in air conditioning or in the high-voltage precipitation, whether it is positive or negative, but it is peculiar to one kind of dust, namely, a dust that has a high resistivity. I would prefer to see this curve plotted, along with the relative humidity, with the electrical resistivity of the dust.

It would be helpful in the interpretation of this curve to have a curve on this chart for clean electrodes. I notice that the curve marked *before* is stated to be one where the dust was blown off the electrodes. I doubt that blowing will give a clean electrode. It is my experience that when dealing with this peculiar kind of dust and electrodes that have been wiped clean, enough dust remains to give the effect noted.

By emphasizing gypsum dust a distorted picture is created as it is very seldom that pure gypsum would be found. By mixing a little bit of some other kind of dust with it the results would be quite different.

It is my experience that Ladenburg's theory is perfectly demonstrable as giving good results provided conditions obtain in the discharge for which Ladenburg derived his formula. For instance, in any kind of a device there are theories which explain its conditions, its operation when it is working under design conditions, but you can usually do something to that machine which introduces new problems. As I see it, this kind of a dust does exactly that.

So I regard Ladenburg's theory as useful and applicable, provided we have the conditions in the ionizer for which his equations were worked out, but with gypsum dust as special considerations. This is a very real problem and certainly if we want to precipitate gypsum or some high-resistivity fly ash, or even some kinds of lint, these problems are very real.

AUTHORS' CLOSURE (J. E. Coolidge): It appears that neither in the paper nor in the oral presentation have we made clear the principal point we are trying to make. The purpose of this paper is to point out that a great deal more basic information must be obtained and fitted into a theory before we can pretend to understand completely the operation of commercial dust precipitators. A secondary purpose of the paper is to give some data which have been obtained on a single kind of dust and to offer a theory which seems to fit the recorded phenomena. It is not contended that the dust chosen is a representative type of dust or a dust which can be substituted for any or all others. We would be the first to admit that different results will be obtained with different dusts.

Dr. Penney comments that he would like to see more information than the data shown in Fig. 3. We agree that the addition of a measurement of the electrical resistivity and the repetition of measurements using clean electrodes would be interesting. This is a restatement of our need for a great deal more information. Dr. Penney further says that his experience with Ladenburg's theory is that it *is perfectly demonstrable*. It is not our object to prove Ladenburg's theory. We merely want to say in the paper that Ladenburg's theory does not fit our results and does not often fit the results nor the conditions of a commercial electrostatic precipitator. However, because published information is so scarce, it would be helpful if Dr. Penney would publish some of the results of his experiments which support the Ladenburg theory. As we stated previously a great deal more information is needed before the behavior of commercial precipitators can be thoroughly understood.

1415

No. 1415

THEORY OF EARTH HEAT EXCHANGERS FOR THE HEAT PUMP

By L. R. INGERSOLL*, F. T. ADLER**, MADISON, WIS., H. J. PLASS†, STANFORD UNIVERSITY, CALIF., AND A. C. INGERSOLL††, MADISON, WIS.

THE EARTH is the most-discussed source of heat for reverse-cycle heating because of availability, temperature, and other considerations. While the small pipe, either relatively isolated or in hairpin loops, is probably the most effective means for use in such heat extraction and is the one which has received most attention, theoretical or experimental, there are a number of other forms of exchangers which should be studied. These include the flat surface and flat plate, the large cavity or "buried cistern," pipes of various sizes, large and small, long and short, and various forms of pipe grids. Most of these have been discussed as possibilities by Kemler[1]. This paper is a brief presentation of the theory applying to these various forms together with a comparative study of their efficiencies, and includes a treatment of some phases of the subject which have hitherto had little discussion.

No one knows better than the practical heat pump engineer the short-comings and inadequacies of the theory in this field. There are at least four difficulties encountered by theoretical treatment at the start. These are: (*a*) lack of knowledge of the thermal conductivity and diffusivity of the soil in the given location; (*b*) the well-recognized fact that there is a moisture migration, from the warmer to the cooler regions in the soil, which improves conductivity near the exchanger (winter operation) and produces more favorable transfer conditions than calculated; (*c*) the underground water movement which obtains a larger heat transfer than would be calculated on a pure conduction basis; and (*d*) the ice formation

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[1] Exponent numerals refer to References.

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

about the pipe or other exchanger, with accompanying release of latent heat. To these items might be added at least two others, such as the periodic temperature variation which occurs to some depth below the ground surface, the effects of cyclic rather than continuous operation of the heat pump, and any possible *boundary layer* effect due to lack of intimate contact of the exchanger with the soil.

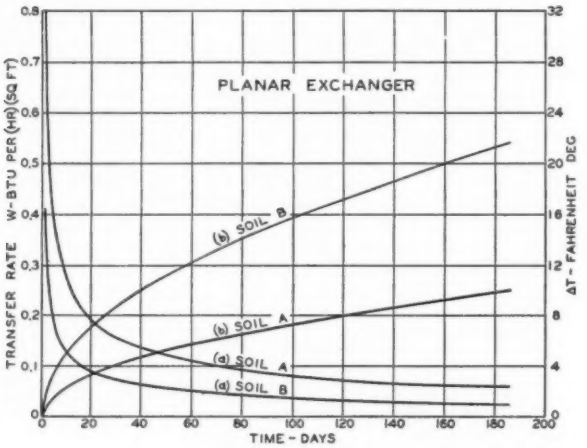

FIG. 1. TRANSFER RATE AT CONSTANT EXCHANGER SURFACE TEMPERATURE, AND TEMPERATURE DIFFERENTIAL AT CONSTANT TRANSFER RATE, FOR PLANAR EXCHANGER[a]

[a] Curves (a) (left-hand scale): transfer rate w for a temperature differential (*i.e.*, temperature difference between surface of exchanger and initial soil temperature) of $\Delta T = 1$ deg. Curves (b) (right-hand scale): temperature differential ΔT for a transfer rate of $w = 1$ Btu per (hr) (sq ft). Soil constants: soil A, $k = 1.2$, $c = 0.45$, $\rho = 102$, $\alpha = 0.0261$; soil B, $k = 0.4$, $c = 0.3$, $\rho = 100$, $\alpha = 0.0133$.

In presenting the accompanying theoretical discussion, the authors recognize these limitations. However, when it comes to a *comparison* of the various types of exchanger, consideration of item (*a*) is largely eliminated since it applies equally to practically all types of earth heat exchangers. Some treatment of items (*b*), (*c*) and (*d*) is attempted in this paper. In further justification of theory it may be added that, while there have been a number of experiments with pipes, it may be some time before there are any experimental comparative studies of all the various types of exchanger; so for the present it is necessary to fall back on theory.

In the accompanying treatment two general cases are considered for each type of exchanger: (1) *constant surface temperature of the exchanger* and the consequent heat transfer as dependent on time, and (2) *constant heat transfer rate* and the resulting surface temperature calculated as a function of time. While the latter would appear to approximate more nearly the situation encountered in practical use—at least it is the heat transfer rate which is generally known or specified—it is frequently easier, experimentally, to meet the former

conditions. The exchangers are treated as isolated in general and remote from ground surface effects, and the initial soil temperature is considered uniform. (For constant transfer rate with a small diameter pipe exchanger any departure from uniformity, such as that due to seasonal changes, can generally be taken into account by merely adding such departure from the mean temperature to the effect, *i.e.*, temperature change, produced by the exchanger.) To take account of variations from these conditions would unduly extend the treatment; furthermore they have been considered in an earlier paper[2] on pipe exchangers, and on the basis of this an estimate of their effect can be reached. Likewise, for present purposes, the somewhat secondary effects of cyclic operation are neglected and only average transfer rates and temperatures are considered. As in the earlier paper, calculations are limited to two types of soil: (1) Soil A, a wet clay of high conductivity having the characteristics $k = 1.2$, $c = 0.45$, $\rho = 102$, $\alpha = 0.0261$ (these are to be considered, of course, only as effective overall values) and (2) Soil B, a moist soil of medium low conductivity ($k = 0.4$, $c = 0.3$, $\rho = 100$, $\alpha = 0.0133$).*

PLANE SURFACE OR FLAT PLATE

Consideration is given here to a plane surface (heat flow perpendicular to the surface and in the positive x direction only), or a flat plate (flow in both positive and negative x directions) which is so large that edge effects can be neglected.

(*a*) *Constant Exchanger Surface Temperature.* This is one of the simplest cases. The equation from ref. 3 p. 90†

$$w = \frac{k\Delta T}{\sqrt{\pi\alpha t}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

is well known and has been ingeniously graphed by Raber, Boester and Hutchinson.[4]

Sample calculations: For a large plane surface in soil A, held at a constant temperature differential ΔT** of 1 deg above or below initial soil temperature, the rate of heat transfer w after 168 hr or one week is 0.323 Btu per (hr) (sq ft) and 0.111 after two months (see Fig. 1). For soil B the corresponding values are 0.151 and 0.051. The values would be the same for a flat plate transferring heat on both sides, but, for the same general dimensions, it would obviously have twice the area of flat surface and consequently twice the total heat transfer.

(*b*) *Constant Transfer Rate.* The equation here is

$$\Delta T = \frac{2w\sqrt{\alpha t}}{k\sqrt{\pi}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

and is based on the flat plate equation[3] (p. 156; the factor of 2 discrepancy occurs because of the way the area is taken in this reference). In using either

* Tables of symbols are included at the end of this paper.

† Page numbers appearing with references refer to page numbers in the reference work.

** The expression *temperature differential* ΔT is used throughout this paper as an abbreviation for $T - T_0$, *i.e.*, the difference between the temperature at the point under consideration (usually the surface of the exchanger) and the *initial* uniform soil temperature. It is obviously constant for case (*a*) and a function of time for case (*b*).

Equations 1 or 2 for a flat plate transferring heat on each side, w is calculated as the total area of the two sides.

Sample calculations: For a large plane surface in soil A, with transfer rate w of one Btu per (hr) (sq ft) (*i.e.*, total transfer of 600 Btu per hr for each 20×30 ft surface, or 1200 for a plate of this size), ΔT would be 1.97 deg after one week and 5.80 deg after two months, *vs.* 4.20 deg and 12.4 deg for soil B (see Fig. 1).

It is to be noted that *this temperature differential is proportional to the square root of the time.* Guernsey, Betz and Skau[5] have made calculations on a long-time (20 year) basis for one soil and have also treated the case of periodic flow.

SPHERICAL CAVITY

The spherical cavity might well be called a *spherical exchanger,* but the term *cavity* expresses the condition better.

(*a*) *Constant Temperature of Cavity Surface.* The equation[3] (p. 143) which applies is

$$q = 4\pi k R \Delta T \left(1 + \frac{R}{\sqrt{\pi \alpha t}} \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Sample calculations: For a cavity of radius $R = 10$ ft and $\Delta T = 1$ deg the transfer rate for soil A is $q = 557$ Btu per hr and $w = 0.443$ Btu per (hr) (sq ft) after one week, and 288 and 0.229 respectively after two months. For soil B the corresponding values are 240 and 0.191; and 114 and 0.091 (see Fig. 2).

(*b*) *Constant Transfer Rate.* It can be shown that the temperature at a point in the medium distant r from the center of a spherical cavity of radius $R (r > R)$ in which heat transfer with the medium is taking place at a constant rate q, differs from its initial value by an amount ΔT which is given by

$$\Delta T = \frac{q}{4\pi k r} \left[\frac{2}{\sqrt{\pi}} \int_{\frac{r-R}{2\sqrt{\alpha t}}}^{\infty} e^{-\beta^2} d\beta - e^{\frac{r-R}{R} + \frac{\alpha t}{R^2}} \frac{2}{\sqrt{\pi}} \int_{\frac{r-R}{2\sqrt{\alpha t}} + \frac{\sqrt{\alpha t}}{R}}^{\infty} e^{-\beta^2} d\beta \right] . \quad (4)$$

When $r = R$, Equation 4 reduces to

$$\Delta T = \frac{q}{4\pi k R} \left[1 - e^{\frac{\alpha t}{R^2}} \cdot \frac{2}{\sqrt{\pi}} \int_{\frac{\sqrt{\alpha t}}{R}}^{\infty} e^{-\beta^2} d\beta \right] \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Sample calculations: For the same cavity as used in (*a*) and for a transfer rate w ($w = q/4\pi R^2$) of one Btu per (hr) (sq ft) of cavity surface the temperature differential ΔT after one week (soil A) is 1.65 deg and after two months 3.65 deg. For soil B the values are 3.70 deg and 8.80 deg (see Fig. 2).

It may be noted that an actual steady state of temperature is eventually attained in the spherical cavity case. (In either the infinite planar exchanger or the long isolated pipe there is no true steady state.) In the example given it would be reached approximately in a few years, when ΔT would be

$$\Delta T = \frac{q}{4\pi k R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

This gives $\Delta T = 8.33$ deg for soil A and 25 deg for soil B. This would mean, from Equation 3, a transfer rate (for $\Delta T = 1$ deg) of $q = 150.8$ Btu per hr, or $w = 0.120$ Btu per (hr) (sq ft) for soil A and corresponding values of 50.2 and 0.040 for B.

These same general considerations apply to any cavity of roughly spherical (or even cubical) shape, *e.g.*, a "buried cistern". As will be shown, the cavity

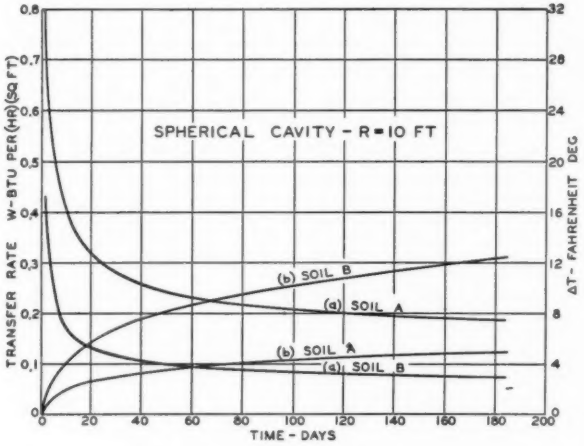

FIG. 2. TRANSFER RATE AT CONSTANT EXCHANGER SURFACE TEMPERATURE, AND TEMPERATURE DIFFERENTIAL AT CONSTANT TRANSFER RATE, FOR SPHERICAL CAVITY EXCHANGER[a]

[a] Curves (a) (left-hand scale): transfer rate w for $\Delta T = 1$ deg. Curves (b) (right-hand scale): temperature differential ΔT for a transfer rate of $w = 1$ Btu per (hr) (sq ft). For soil constants see Fig. 1.

(also the flat plate) is a relatively inefficient exchanger. The presence of corners and edges, as in the cubical shape, does not increase its efficiency markedly; this may be estimated by comparing its efficiency in steady state operation with that of a sphere of equal volume (see Polya[6]).

PIPES

(*a*) *Constant Surface Temperature.* The theory of this case has been worked out by Jaeger[7] and others (see also Carslaw and Jaeger[8] p. 283). The rate of heat exchange Q' Btu per (hr) (linear ft) for a long isolated pipe held at a constant temperature ΔT above or below the initial soil temperature is given by

$$Q' = k \Delta T F(z) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the rate w, Btu per (hr) (sq ft) by

$$w = \frac{k \Delta T}{2\pi R} F(z) \quad \ldots \ldots \ldots \ldots \quad (7)$$

where $z \equiv \alpha t/R^2$ and

$$F(z) \equiv \frac{8}{\pi} \int_0^\infty \frac{e^{-z\beta^2}}{J_0^2(\beta) + Y_0^2(\beta)} \cdot \frac{d\beta}{\beta} \quad \ldots \ldots \ldots \quad (8)$$

Values of $F(z)$, adapted and extended from Jaeger and Clarke[9], are given in Table 1.

Sample calculations: For a 4 in. (O.D.) pipe* (*i.e.*, $R = 0.167$ ft) in soil A, with $t = 24$ hr and $\Delta T = 1$, values of z and $F(z)$ will be: $z = 22.5$ and $F(z) = 2.83$. Then $Q' = 3.40$

TABLE 1—VALUES OF $F(z)$ (SEE EQUATION 8)

z	$F(z)$	z	$F(z)$	z	$F(z)$	z	$F(z)$
0.01	38.508	0.50	7.751	40	2.539	4000	1.350
0.02	28.096	0.60	7.288	50	2.440	5000	1.319
0.03	23.474	0.70	6.926	60	2.363	7500	1.266
0.04	20.716	0.80	6.634	80	2.251	10000	1.233
0.05	18.829	1.00	6.180	100	2.172	15000	1.184
0.06	17.433	1.2	5.844	120	2.114	25000	1.131
0.07	16.351	1.5	5.467	150	2.042		
0.08	15.472	2.0	5.029	200	1.956		
0.09	14.744	2.5	4.726	250	1.893		
0.10	14.130	3.0	4.500	300	1.846		
0.12	13.135	4	4.174	400	1.775		
0.14	12.361	5	3.947	500	1.721		
0.16	11.737	6	3.776	600	1.683		
0.18	11.217	8	3.529	800	1.622		
0.20	10.777	10	3.354	1000	1.578		
0.25	9.916	12	3.224	1200	1.543		
0.30	9.277	15	3.074	1500	1.502		
0.35	8.779	20	2.898	2000	1.454		
0.40	8.373	25	2.774	2500	1.418		
0.45	8.037	30	2.678	3000	1.393		

Btu per (hr) (linear ft), or $w = 3.23$ Btu per (hr) (sq ft). For $t = 731$ hr, $Q' = 1.98$ and $w = 1.88$ (see Fig. 3).

(*b*) *Constant Transfer Rate.* The simplest theory for this is the line source equation[2],

$$\Delta T = \frac{Q'}{2\pi k} \int_{\frac{r}{2\sqrt{\alpha t}}}^\infty \frac{e^{-\beta^2}}{\beta} d\beta \equiv \frac{Q'}{2\pi k} I\left(\frac{r}{2\sqrt{\alpha t}}\right) \quad \ldots \ldots \quad (9)$$

but in cases where the time is too short or the pipe diameter too large (in general when $\alpha t/R^2 < 20$) the use of the line source equation involves appreciable error. Jaeger[7] has given an equation, which, while difficult to evaluate, covers all cases (see also ref 8 p. 283). For a pipe transferring heat at a rate of Q'

* For simplicity and because of the variety of materials and sizes used in exchanger pipes, diameters are expressed in whole numbers of inches, rather than standard pipe sizes.

Btu per (hr) (linear ft) the temperature differential ΔT between the initial soil temperature and the point under consideration at or near the pipe is given by

$$\Delta T = \frac{Q'}{k} G(z, p) = \frac{2\pi R w}{k} G(z, p) \quad \ldots\ldots\ldots (10)$$

where

$$G(z, p) \equiv \frac{1}{\pi^2} \int_0^\infty \frac{e^{-\beta^2 z} - 1}{J_1^2(\beta) + Y_1^2(\beta)} \left[J_0(p\beta)\, Y_1(\beta) - J_1(\beta)\, Y_0(p\beta) \right] \frac{d\beta}{\beta^2} \quad \ldots (11)$$

In these equations $z \equiv \alpha t/R^2$ as above, and $p \equiv r/R$; *i.e.*, for $p = 5$ the point whose temperature is sought is distant $5R$ from the center of the pipe. The

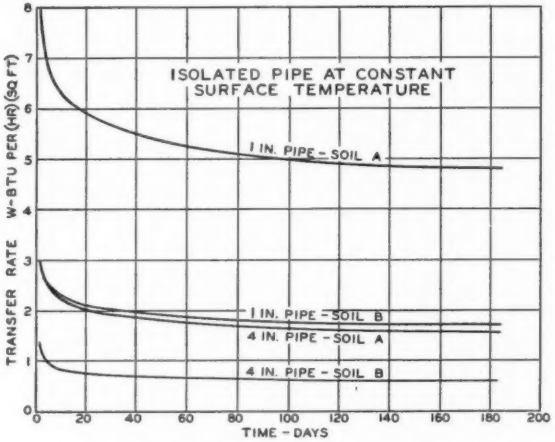

FIG. 3. TRANSFER RATE FOR LONG ISOLATED PIPE EXCHANGER AT CONSTANT SURFACE TEMPERATURE[a]

[a] Temperature differential $\Delta T = 1$ deg. For soil constants see Fig. 1.

values for $p = 1$ apply to the pipe surface. The values of $G(z,p)$ have been computed for a wide range of values of z (see ref. 8, p. 283 for values up to $z = 10$ for $p = 1$) and for $p = 1,2,5,10$. These are given in Table 2.

Sample calculations: For the same 4 in. O. D. pipe and soil A used in calculations under previous item (a), with $Q' = 1$ Btu per (hr) (linear ft) and $t = 5.34$ hr, values of z and $G(z, 1)$ will be $z = 5$ and $G(z, 1) = 0.217$. The pipe surface temperature change would accordingly be $\Delta T = 0.181$ deg. For $w = 1$ this figure would be 0.190 deg (see Figs. 4 and 5). For a distance of $5 \times 0.167 = 0.833$ ft from the center of the pipe, $\Delta T = 0.0127$ deg for $Q' = 1$.

In Fig. 5 are plotted curves calculated on the basis of the line source equation (Equation 9) and also of the more exact Jaeger integral (Equation 11) evaluated in Table 2. It is to be particularly noted that, with ordinary soils and *for pipes less than 4 in. in diameter and times longer than one day, the simple line source equation* (see ref. 2, or ref. 3, p. 253, for tables of this integral) *gives*

results suffiently accurate for practical purposes. The error does not exceed about 2 percent under these conditions. More exactly, the criterion for this small error or less is $\alpha t/R^2 > 20$.

This criterion also holds in general for the use of the line source equation for points at a distance from the pipe, R still being the radius of the *pipe*. This

TABLE 2—VALUES OF $G(z,p)$ (SEE EQUATION 11)

z	G (z, 1)	G (z, 2)	G (z, 5)	G (z, 10)
0.10	0.049			
0.20	0.067			
0.30	0.080			
0.40	0.090			
0.50	0.099			
0.60	0.107			
0.70	0.113			
0.80	0.118			
1.00	0.128	0.035	0.001	0.000
1.20	0.137			
1.5	0.148			
2.0	0.163			
2.5	0.175			
3.0	0.186			
4.0	0.203			
5	0.217	0.112	0.0153	0.0001
6	0.228			
8	0.247			
10	0.263	0.155	0.0388	0.0024
12	0.275	0.167	0.0470	0.0042
15	0.291	0.182	0.0580	0.0072
20	0.312	0.203	0.0736	0.0129
25	0.328	0.219	0.0866	0.0188
30	0.342	0.232	0.0979	0.0246
50	0.380	0.271	0.132	0.0460
100	0.433	0.323	0.181	0.0842
500	0.560	0.449	0.304	0.197
1000	0.614	0.504	0.359	0.250
5000	0.742	0.632	0.486	0.376
10000	0.797	0.687	0.541	0.431
25000	0.870	0.760	0.614	0.504

equation can therefore be used to find the temperature 20 in. from the center of a 4 in. pipe in soil A after 24 hr (*i.e.*, $\alpha t/R^2 = 22.5$) but the Jaeger integral would be needed for an 8 in. pipe for this distance and time. For $r > 10R$ the Jaeger integral (Equation 11) is not evaluated, so the line source equation is the only simple solution readily available without further numerical work (the limits of its validity to be kept in mind).

(*c*) *Short Pipes.* The preceding theory applies exactly only to the infinitely long pipe. It is important accordingly to know something of the magnitude of the error involved when it is applied to pipe lengths in common use.

The theory of the finite pipe for the case of constant transfer rate is not hard to derive. The temperature differential at a distance r from a point source at time t after it starts to emit or absorb q units of heat per unit of time is given (ref. 3, p. 144, 146) by

$$\Delta T = \frac{q}{4\pi k r} \cdot \frac{2}{\sqrt{\pi}} \int_{\frac{r}{2\sqrt{\alpha t}}}^{\infty} e^{-\beta^2} d\beta \quad \ldots\ldots\ldots \quad (12)$$

For a line source in the z direction made up of such points, emitting Q' heat units per unit of time per unit of length, the temperature differential at a distance R (*e.g.*, on the pipe surface) from this line is

$$\Delta T = \frac{Q'}{4\pi k} \int_{z_1}^{z_2} \frac{dz}{\sqrt{z^2 + R^2}} \int_{\frac{\sqrt{z^2+R^2}}{2\sqrt{\alpha t}}}^{\infty} \frac{2}{\sqrt{\pi}} e^{-\beta^2} d\beta \quad \ldots\ldots \quad (13)$$

where the pipe extends from z_1 to z_2. When applied to pipe surface temperatures, this expression, like the line source equation, is reasonably accurate only for $\alpha t/R^2 > 20$, as would be true for most practical cases.

For $(\sqrt{z^2+R^2}/2\sqrt{\alpha t}) < 1$, Equation 13 can be put in the form

$$\Delta T = \frac{Q'}{4\pi k}\left[\log_e \frac{z_2 + \sqrt{z_2^2 + R^2}}{z_1 + \sqrt{z_1^2 + R^2}} - \frac{1}{\sqrt{\pi \alpha t}} \left\{ \left(1 - \frac{R^2}{12\alpha t} + \frac{R^4}{160\alpha^2 t^2} - \ldots. \right) \left(z_2 - z_1 \right) + \left(-\frac{1}{12\alpha t} + \frac{2R^2}{160\alpha^2 t^2} - \ldots. \right) \left(z_2^3 - z_1^3 \right) \Big/ 3 + \left(\frac{1}{160\alpha^2 t^2} + \ldots. \right) \left(z_2^5 - z_1^5 \right) \Big/ 5 + \ldots. \right\} \right] \quad \ldots\ldots\ldots\ldots \quad (14)$$

For $R < L_p/10$ and $L_p < \sqrt{\alpha t}$ * (L_p = pipe length) as would frequently be the case, Equation 14 reduces to the following simple design equation for pipe surface temperatures near the middle of a short pipe,

$$\Delta T = \frac{Q'}{4\pi k}\left[4.6 \log_{10} \frac{L_p}{R} - \frac{L_p}{\sqrt{\pi \alpha t}} \right] \quad \ldots\ldots\ldots \quad (15)$$

This gives, for example, the temperature at the surface in the middle of a 2 in. pipe 4 ft long in soil A, after a few days of operation, with negligible error.

While there is no steady state of temperature for the infinite isolated pipe there is such a state for the finite pipe—although it may take a long time to

---

* This is for an error of about 1 percent. For $L_p = 2\sqrt{\alpha t}$ the error is of the order of 4 percent, and for $L_p = 3\sqrt{\alpha t}$, about 9 percent.

reach this state—and its value is given at once by putting $t = \infty$ in Equation 13 whence

$$\Delta T = \frac{Q'}{4\pi k} \int_{z_1}^{z_2} \frac{dz}{\sqrt{z^2 + R^2}} = \frac{Q'}{4\pi k} \log_e \frac{z_2 + \sqrt{z_2^2 + R^2}}{z_1 + \sqrt{z_1^2 + R^2}} \quad . . (16)$$

The curves of Fig. 6 show the results of such calculations made either by graphical solution of Equations 13 and 16 or by the use of Equations 14 or 15, for points on the surface of a 2 in. pipe ($R = 0.0833$ ft) for soils A and B. It will be noted (curves I or L) that for the 40 ft pipe (except for the last foot of pipe)

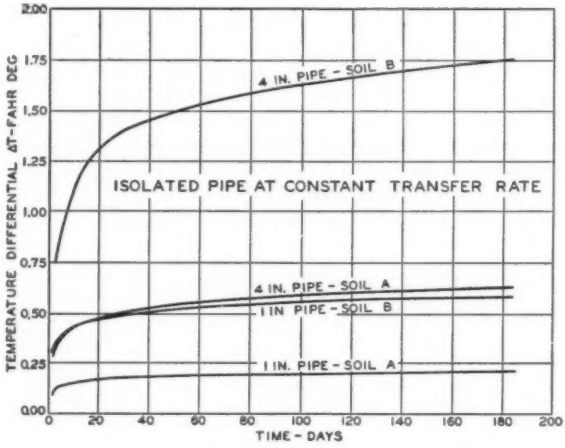

FIG. 4. TEMPERATURE DIFFERENTIAL FOR LONG ISOLATED PIPE EXCHANGER AT CONSTANT TRANSFER RATE[a]

[a] Transfer rate $w = 1$ Btu per (hr) (sq ft). For soil constants see Fig. 1.

the temperatures are practically the same as for the infinite pipe (curves K or N). This means that *the infinite pipe theory may in general be applied with negligible error to the long pipes usually used.* It is also to be noted that while the 4 ft long pipe comes within 20 percent of steady state temperatures after one week of operation, the 40 ft pipe would require a very much longer time for such attainment.

With the aid of another integration, Equation 13 can be modified to apply to the case of the small flat plate exchanger.

While there is no steady state for the isolated infinite pipe, there is, however, a steady state for an infinite pipe, *with constant transfer rate,* buried below a ground surface held at constant temperature. If the ground surface temperature is the initial soil temperature, the use of Equation 9 gives this steady state temperature at once. The value of this integral for large t is simply $\log_e 2\sqrt{\alpha t}/r$

(see ref. 3, p. 253) so for a pipe buried a distance s below the surface, ΔT (if the radius of the pipe is small relative to s) for the steady state, is

$$\Delta T = \frac{Q'}{2\pi k}\left(\log_e \frac{2\sqrt{\alpha t}}{r} - \log_e \frac{2\sqrt{\alpha t}}{2s}\right) = \frac{Q'}{2\pi k}\log_e \frac{2s}{r} \quad . \quad . \quad . \quad (16a)$$

Equation 16*a* is frequently used, especially for buried cables, but its derivation by the usual method is by no means simple (see ref. 10, pp. 333-4).

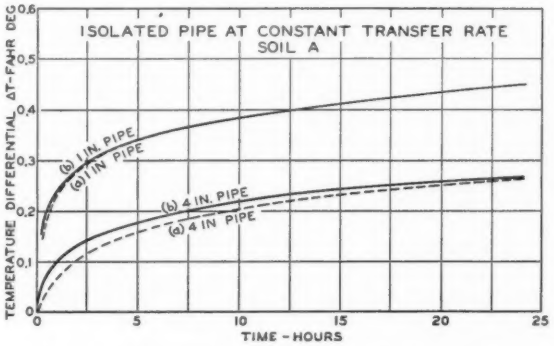

FIG. 5. COMPARISON OF RESULTS OF COMPUTATIONS FROM THE LINE SOURCE EQUATION AND THE JAEGER INTEGRALS OF THE TEMPERATURE DIFFERENTIAL FOR LONG ISOLATED PIPE EXCHANGER AT CONSTANT TRANSFER RATE[a]

[a] Transfer rate $Q' = 1$ Btu per (hr) (linear ft). The approximate curves (a) (dashed line) are obtained with the line source Equation 9; the exact curves (b) are computed from Table 2 of the Jaeger integrals. For soil constants see Fig. 1.

COMPARISON OF EXCHANGERS

For comparison purposes, the values of w (the transfer rate per square foot of area) for the planar exchanger, spherical cavity, and two sizes of pipe have been plotted together in Fig. 7. The superiority of the pipe exchanger is at once evident, not only because its transfer rate is large but also because the reduction of heat transfer rate with time is less. It is also to be noted that the small pipe has a higher transfer rate than the larger one.

The comparisons of Fig. 7 are all on the basis of the transfer rate *per square foot of area* of the various exchangers. If it is desirable to compare the two sizes of pipe on the basis of the transfer rate Q' *per linear foot*, it is only necessary to multiply the ordinates by π times the pipe diameter. From this it may be concluded that Q' for the 4 in. pipe is from 1.29 (soil A, 6 months) to 1.68 (soil B, 1 day) larger than for the 1 in. pipe. This indicates the desirability of backfilling around a small pipe in a trench with some material of as high

conductivity as possible. It also goes without saying that close adherence of the soil to the pipe is of the highest importance.

PIPE GRIDS

Most exchangers in actual use or under test are pipe grids of various types. Exact calculations are difficult in most cases but some general observations can be made.

(*a*) *Hairpin Loops.* These have been treated in the previous paper[2]. The curves show that the single loop still has a good efficiency per pipe. Applications of the

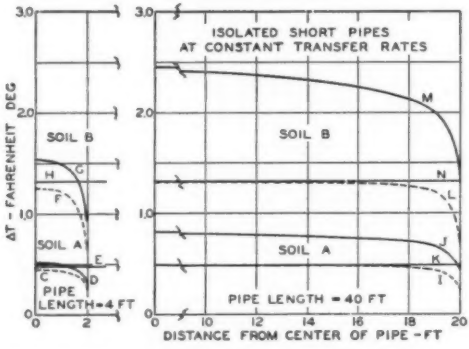

FIG. 6. TEMPERATURE DIFFERENTIAL FOR ISOLATED SHORT PIPE EXCHANGERS AT CONSTANT TRANSFER RATE[a]

[a] Transfer rate $Q' = 1$ Btu per (hr) (linear ft). Diameter of pipe, 2 in. Length, 4 ft and 40 ft (with infinite pipe for comparison). Curves give temperature differential for points on pipe surface at various distances from middle of pipe for the following conditions: C, time 168 hr; D, infinite time, *i.e.*, steady state; E (straight line), infinite pipe, 168 hr; F, time 168 hr; G, steady state; H, infinite pipe, 168 hr; I, time 168 hr; J, steady state; K, infinite pipe (same as E); L, time 168 hr; M, steady state; N, infinite pipe (same as H). For soil constants see Fig. 1.

same equations, however, will show that with the addition of more pipes to the same trench the efficiency falls off markedly. This is well shown in Fig. 8 where the temperature differential is plotted for an isolated 1 in. pipe, for each pipe in a hairpin loop with the pipes 30 in. apart, and for each pipe in two such loops 30 in. apart. In the latter case each pipe is 30 in. distant from two similar pipes absorbing heat at 1 Btu per (hr) (linear ft) and 42.4 in. from a third. The temperature of any one pipe can accordingly be calculated as in the previous paper[2] as the sum of the effects of line sources at distances of 0.5 in., 30 in., 30 in., and 42.4 in.

(*b*) *Flat grids.* A closely-spaced (perhaps a foot or so between pipes) flat grid becomes effectively a flat plate, whose treatment has already been given. It has occasionally been suggested that pipes might be wound about the foundation of a house (with suitable insulation next to the wall) or located under the basement floor. Equations 1 and 2 would show that such a grid is relatively inefficient and it is, moreover, open to special serious practical objections based on freezing of ground.

(*c*) *Star Grids.* In such a grid, the loops, each more or less in a vertical plane, radiate out like the points of a star from a central header. Success could not be predicted for such a grid because of the small volume of soil with which it is concerned. Such reports as are available indicate that such grids are unsatisfactory.

(*d*) *Pipe Helixes.* A closely-wound helix becomes, effectively, a pipe of the diameter of the helix. The considerations brought out in the previous section would indicate that this is likely to be a waste of pipe and little more effective than a single pipe of the same overall length and diameter as the helix.

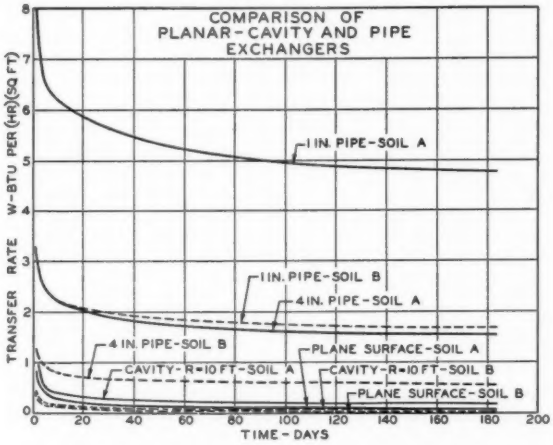

FIG. 7. COMPARISON OF TRANSFER RATES FOR PLANAR, SPHERICAL CAVITY, AND LONG ISOLATED PIPE EXCHANGERS AT CONSTANT SURFACE TEMPERATURE[a]

[a] Temperature differential $\Delta T = 1$ deg. For soil constants see Fig. 1.

(*e*) *Pipe with Laterals.* Since the end of a pipe operates at a smaller temperature differential (see Fig. 6) for a given transfer rate than the rest of the pipe, a long pipe with a large number of short (*e.g.*, 3 ft) laterals might have possibilities.

(*f*) Vertical *Pipe Grid.* Wagley[11] and Randolph have been testing§ an ingenious grid made up of some forty 10 ft lengths of 3 in. (3.5 in. O.D.) iron pipe installed vertically with the aid of a pole-setting digger. These were arranged in four rows of 10 pipes each, the alternate rows being staggered, with a spacing between the pipes of 9 ft 3 in. This meant that, except at the ends of the rows, each pipe in the two inside rows was at a distance of 9 ft 3 in. from each of six similar pipes. The plot of ground used was approximately 40 × 75 ft.

A series of measurements had been made with thermocouples at different depths in the ground before the pipes were installed. These gave an annual temperature range of 34 deg at a depth of 2 ft and 4 deg at 22 ft. From these measurements, using the equation[3] (p. 47),

$$T_R = 2T_0 e^{-x\sqrt{\pi/\alpha P}} \quad \ldots\ldots\ldots\ldots \quad (17)$$

§ Results of tests used by permission.

which expresses the range at any depth x in terms of the surface range $2T_o$ (T_o = amplitude or half temperature range at surface) the diffusivity α, and the period P of 8766 hr (one year), the diffusivity of this soil may be calculated as 0.028. (It is unnecessary to know the surface range because measurements were taken at two different depths). Assuming average values for c as 0.30 and ρ as 105 the value of k is 0.88. Calculations based on temperature lag for different depths give practically the same value for α. These values will therefore be used as the thermal constants of this soil.

In a recent all-winter test heat was extracted at an average rate of about 35 Btu per (hr) (foot length of the 3.5 in. pipe) for a period of some months, and in 3

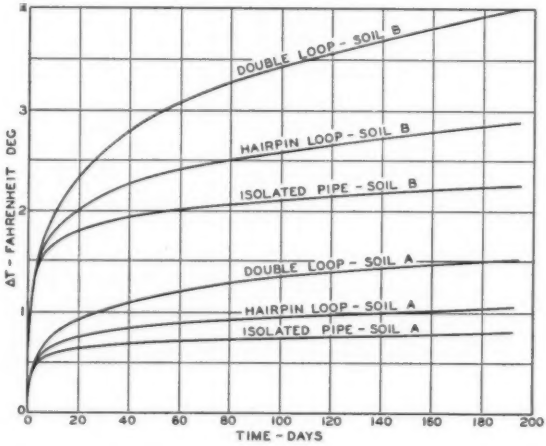

FIG. 8. TEMPERATURE DIFFERENTIAL FOR ISOLATED PIPE, HAIRPIN LOOP, AND DOUBLE LOOP AT CONSTANT TRANSFER RATE[a]

[a] Transfer rate $Q' = 1$ Btu per (hr) (linear ft). Pipes 1 in. diameter, 30 in. apart in loop and the two loops 30 in. apart.

months this resulted in a 30 deg drop at the pipe surface, bringing it several degrees below freezing. Using the given soil constants, the temperature differential for an isolated pipe of this size is at once calculated from Table 2 and Equation 10 (or from Equation 9 if desired) as 27.7 deg. However, since each pipe in this test was only 9.25 ft distant from 3.7 (average) other pipes, calculations (Equation 9 must be used here) give an additional drop of 9.3 deg making a total drop of 37 deg. When it is remembered that no account has been taken of the latent heat released when the freezing point is reached, also that moisture migration in the soil will improve the conductivity near the pipes and hence lessen the temperature drop, it must be considered that the calculation is in satisfactory agreement with the measured value of 30 deg.

In the tests reported so far only half of the total grid of 40 pipes has been used. When all are in operation at this same rate a few degrees greater temperature drop may result from the proximity, for any one pipe, of more neighboring pipes.

GROUND TEMPERATURE RECOVERY

When a large block of soil is chilled by heat extraction in winter some 30 deg or more below its original temperature, as in the installation just described, the question of its temperature recovery in summer becomes an important one. Some approximate calculations may be made assuming that such recovery will result from the heat inflow at the ground surface and from the surrounding soil, and neglecting any heat returned as a result of summer operation.

A block of soil is assumed to be 40×75 ft in area and 14 ft deep, and having the thermal conductivity constants ($k = 0.88$, $\alpha = 0.028$), chilled to ΔT below the initial soil temperature (which is perhaps 50 F) and in contact with soil at this original temperature. Assume also that the upper surface is held at this 50 F temperature. The temperature recovery which might be expected in five months is desired at a point in the middle of the long dimension of the block 3 ft from the longitudinal center line and 10 ft below the surface.

The equation which applies here[3] (p. 181) is

$$\frac{\Delta' T}{\Delta T} = \frac{1}{8}\left(\frac{2}{\sqrt{\pi}}\right)^3 \int_{-37.5p}^{37.5p} \int_{-23p}^{17p} \left[\int_{-4p}^{10p} - \int_{10p}^{24p} \right] e^{-\beta^2 - \gamma^2 - \epsilon^2} \, d\beta \, d\gamma \, d\epsilon \quad . \quad (18)$$

Here ΔT is the maximum temperature differential or maximum departure from the original 50 F and $\Delta' T$ is the departure after 5 months of recovery. The quantity $p \; 1/2\sqrt{\alpha t} = 0.0494$. This equation gives on solution (which is not as difficult as it looks), $\Delta' T/\Delta T = 0.14$. This means that the final departure from the original 50 F is only 14 percent of the maximum departure of say 30 deg, or a recovery of 86 percent. Other points in the block would, in general, show a better recovery, which might average 90 percent or better for the block as a whole. For a block $20 \times 75 \times 14$ ft the central point (7 ft deep) would show a 93 percent recovery. As a matter of fact, recent measurements[11] in such a block indicate this relatively complete recovery.

If the 400 ft of 3 in. pipe had been installed in 20 ft lengths in a block $20 \times 75 \times 24$ ft deep, the recovery would not have been quite so satisfactory. For a central point 16 ft below the surface it might have been 82 percent. For a continuous (*i.e.*, infinite) slab 24 ft thick the recovery for such a point would be about 65 percent.

The matter of summer temperature recovery has an important bearing on the optimum pipe length for a grid of this sort. Since the surface is the most important factor in the recovery, a block of area larger than, say, 60×60 ft should not go much deeper than 24 ft. Long narrow blocks would show better recovery than square ones, *i.e.*, a block 30×120 ft would be better than one 60×60 ft. If any one horizontal dimension is longer than about 60 ft it is immaterial how long it is; *i.e.*, blocks of 30×70 and 30×500 ft would show about equal recovery.

In this discussion there has been assumed a sharp initial temperature discontinuity at the edges of the block at the beginning of the 5 month recovery period. This, of course, would be far from the real case. The actual recovery, taking this into account, would be a little less than that calculated, but the comparisons would still hold. It is also to be noted that summer ground surface

temperatures would generally run higher than the 50 F assumed here, making for more complete recovery.

EFFECT OF ICE FORMATION

The theory here is somewhat involved and will be reproduced only in bare outline. Following the general reasoning of the Neumann ice formation treatment[3] (p. 191) results in the equation

$$k_1 \left(\frac{\partial T_1}{\partial r}\right)_{r=R} - k_2 \left(\frac{\partial T_2}{\partial r}\right)_{r=R} = L \frac{dR}{dt} \quad \ldots\ldots \quad (19)$$

which expresses the fact that at the surface of the cylinder of frozen soil of radius R about the pipe, the difference between the rates of heat flow from this cylindrical boundary towards the center and into this region from the surrounding soil, is the rate of release of latent heat. L is the heat of fusion per unit volume of the soil.

Combining this reasoning with that of the line source Equation 9, it can be shown that

$$R = 2\sqrt{bt} \quad \ldots\ldots\ldots\ldots\ldots \quad (20)$$

where b is a constant. Also that

$$\frac{Q'}{2\pi} e^{-b/\alpha_1} - k_2 \frac{T_0 - T_F}{I\,(b/\alpha_2)} e^{-b/\alpha_2} = 2\,L\,b \quad \ldots\ldots\ldots \quad (21)$$

and

$$T_1 = T_F - \frac{Q'}{2\pi k_1} I\left(\sqrt{b/\alpha_1}\right) \left[\frac{I\left(\frac{r}{2\sqrt{\alpha_1 t}}\right)}{I\left(\sqrt{\frac{b}{\alpha_1}}\right)} - 1 \right] 0 < r \leq R \quad \ldots \quad (22)$$

where k_1, α_1 apply to frozen and k_2, α_2 to unfrozen soil. T_0 is the initial soil temperature, taken as 50 F, T_1 the temperature at any radius r inside the frozen cylinder, and T_f the freezing temperature. I is the line source integral in Equation 9.

In illustrating the use of these equations it is assumed that a cylinder of 1 ft radius (*i.e.*, $R = 1$) of frozen soil is formed about the pipe in 6 months (4383 hr) of operation in soil A, with a heat of fusion of 3672 Btu per cu ft (*i.e*, about 25 percent moisture). Equation 20 then gives $b = 5.70 \times 10^{-5}$. Substituting for b in Equation 21 and assuming that $\alpha_1 = \alpha_2 = 0.0261$ and $k_1 = k_2 = 1.2$, $Q' =$ 51.4 Btu per (hr) (linear ft) under these conditions. Without freezing, the use of the simple line source equation shows that to produce in soil A, initially at 50 F, a temperature of 32 F at $r = 1$ ft in 4383 hr requires that Q' be 48.8 Btu per (hr) (linear ft).

There is a simple way of checking this result. Neglecting the relatively small volume of the pipe, the volume of the frozen soil per foot of pipe for $R = 1$ is 3.14 cu ft, giving a total heat release for absorption by the pipe of 11,540 Btu. If released at a uniform rate over 6 months this would give 2.6 Btu per (hr) (linear ft) as the contribution due to freezing, which is just the difference between the two results calculated above.

Perhaps the greatest interest centers about calculations of the pipe surface temperature with and without freezing considerations. Using Equation 22 with a 2 in. pipe ($r = 0.0833$ ft), and with $Q' = 51.4$, $T_f = 32$ F, and b as calculated above, will give $T = 15.1$ F. Neglecting freezing considerations and using the line source Equation 9 gives $\Delta T = 35.9$ below the initial 50 F, giving a pipe temperature of 14.1 F.

An independent check on this temperature of 15.1 F is obtained by remembering that of the 51.4 absorption rate assumed here only 48.8 comes from the soil, the remainder being furnished by the heat of fusion. It would then be expected that if $Q' = 48.8$ is used in Equation 9, a temperature not far different from 15.1 F should be obtained. As a matter of fact, this procedure gives 15.8 F.

It may be concluded, then, that freezing considerations make relatively little difference in such pipe temperature calculations. As will be shown later it is probable that the moisture migration towards the cold pipe is a larger factor than the ice formation. It is also probable that the frozen soil has a slightly higher conductivity than the unfrozen and that this should enter into the calculations. This is in agreement with the conclusions of Coogan[12].

EFFECT OF MOISTURE MIGRATION

As already mentioned, it is well known that there is a diffusion of moisture from warmer to cooler regions in the soil, perhaps due to vapor pressure differences. This increases the moisture content near a heat-absorbing pipe, with some increase, in general, in thermal conductivity. The diffusivity would not be expected to change as much since additional moisture would increase the specific heat as well as the conductivity. A rough calculation of the effect of such moisture increase on pipe temperatures will be attempted.

The expression for the temperature gradient near a line source or a small pipe is[3] (p. 147),

$$\frac{\partial T}{\partial r} = -\frac{Q'}{2\pi k r} e^{-r^2/4\alpha t} \quad \ldots\ldots\ldots\ldots \quad (23)$$

If this gradient is plotted as a function of r and integrated to obtain the area, the result is an expression which reduces at once to the line source integral of Equation 9. In other words the area under the curve gives the temperature differential. Accordingly, if it is assumed that the conductivity only is changed by the moisture increase and that the diffusivity is relatively constant (in any case it is a somewhat secondary factor), the new gradient may also be plotted by reducing the original gradient by the ratio of the old and new conductivities. The area under the new curve gives the new temperature differential for a pipe absorbing heat at the same constant rate Q'.

Fig. 9 illustrates this. The solid line is a plot of the gradient at the end of six months for a 2 in. pipe in soil A where $Q' = 1$. The area under this curve gives $\Delta T = 0.70$ F at the surface of the pipe, which is just the value computed from Equation 9. If it is assumed that moisture migration doubles the conductivity at the pipe, increases it by 50 percent at a distance of 1 ft and is of negligible effect beyond 5 ft, the dotted curve is obtained and the area under it indicates a ΔT at the pipe of 0.54 F.

When considering the influence of one pipe on another as in neighboring loops, the effect of moisture migration will be secondary. The time required to reach

the assumed moisture densities is also a factor to be considered, although it is believed that it will not greatly alter the final temperatures.

EFFECT OF UNDERGROUND WATER MOVEMENT

It is probable that for a horizontal pipe buried 6 ft or so underground and with a layer of frozen soil above, as in winter operation, underground water movement rarely comes into account. For vertical pipes and greater depths, however, there may be a slow drift of water which may vary from negligibly small

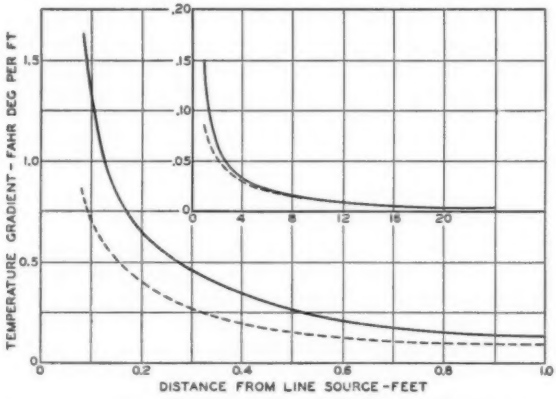

FIG. 9. EFFECT OF INCREASED CONDUCTIVITY, DUE TO MOISTURE MIGRATION, ON THE TEMPERATURE GRADIENT IN THE SOIL NEAR A LINE SOURCE[a]

[a] Transfer rate $Q' = 1$ Btu per (hr) (linear ft). Solid line: Temperature gradient near a line source, i.e., small pipe, in soil A. Dotted line temperature gradient in same soil with conductivity near pipe increased by moisture migration.

values to several feet per day. Because of low velocities and small interstices between the soil grains this flow is non-turbulent.[13]

If measurements by the Slichter[14] or other method indicate such water movement, it is of interest to know about what effect this might be expected to have on the temperature of a pipe exchanging heat at a given rate. Anything like an exact calculation would be very difficult, but some idea of the order of magnitude for the very small velocities most frequently encountered may be reached in two different ways.

The first is by the use of a treatment[8] (p. 224, Equation 3) based on relative motion between the heat source and the medium, a method which would probably give too large values for the effect. Graphical evaluation of the equation for a flow velocity of 0.01 ft per hr in soil A indicates that, after a few months of operation, ΔT for a 2 in. pipe would be 20 percent less than for the same pipe with no water movement. Calculations for higher values of the velocity are much less certain but they indicate that for 0.1 ft per hr the reduction might be by a factor of 5.

In the other method a 2 in. pipe is assumed to be extracting heat from the soil at a rate Q' of 40 Btu per (hr) (linear ft) for two months, giving a certain easily-calculated temperature distribution in the soil about the pipe (soil A, with the assumption that 30 percent of its weight is water). Now let the water flow slowly through this region, adding as it does so, a certain amount of heat to each zone about the pipe. This will give a larger Q' for a given ΔT than would otherwise be the case. Calculations indicate for a flow velocity of 0.01 ft per hr an increase of 19 percent in Q' and for 0.1 ft per hr, 79 percent. While the discrepancy between the results obtained by the two methods of calculations is large for the higher velocity, the results for the other are in good agreement. It may be concluded that water movement gives an effect of the order of 20 percent for a velocity of 0.01 ft per hr and is relatively negligible for much smaller velocities.

SOIL CONDUCTIVITY MEASUREMENTS

It is obviously desirable to be able to determine the thermal conductivity and diffusivity of the soil in a given location. To be of value this should be done with an absolute minimum of soil disturbance because of the effect of any digging up and repacking in lowering the conductivity.

Perhaps the best method is that mentioned in (*f*) of the previous section on pipe grids whereby thermocouples give the temperature at different depths throughout at least one whole year. To meet the need for quicker action the *British Electrical and Allied Industries Research Association*[15] has developed a method using a buried copper sphere 9 in. in diameter (they have also used a cylinder), to which electrical energy is supplied until a steady state is reached. Measurement of its surface temperature then allows (by Equation 5*a*) a calculation of the soil conductivity and simple determinations of specific heat and density then give the diffusivity. The objections to this method are the necessary soil disturbances, the time required for the attainment of the steady state (this might perhaps be shortened by the use of Equation 5), and—most important of all—the fact that the soil would be somewhat dried by the heating and accordingly might give lower than normal values of the conductivity.

To obviate some of these objections it is suggested that a cylindrical slightly tapered calorimeter be used with *guard rings* above and below, that it be kept at a *lower* temperature than the surrounding soil (using melting ice as a measure of the heat abstracted), and that the transient rather than the steady state be used, applying in the calculations the foregoing Table 1. The authors hope to give this method a trial in the near future.

CONCLUSIONS

1. Planar exchangers and roughly spherical cavities are both inferior to the relatively isolated pipe, on the basis of exchanger rate per unit area.

2. On this same basis a small pipe is more efficient than larger ones.

3. In the matter of pipe grids, such forms as flat grids, star grids, and closely-wound pipe helixes are all of questionable efficiency. A hairpin loop with the two parts well apart in a trench should show good efficiency, which would be lowered, however, for two such loops in the same trench. The long pipe with many short laterals is worth consideration and the grid made up of vertical well-separated pipes has good possibilities.

4. Lack of ground temperature recovery in summer need not be a concern unless a large block of ground deeper than some 20 ft has been cooled by winter operation.

5. Freezing of moist soil about the exchanger pipe will retard the cooling of the pipe somewhat, but the effect is not large, being of the order of approximately 10 percent, which accordingly will increase only slightly the efficiency of winter operation.

6. The effect of moisture migration to the cold pipe will increase the soil conductivity and accordingly lessen the temperature gradient near the pipe and therefore lessen the pipe temperature differential. This would tend to increase heat pump efficiency.

7. Underground water movement becomes of notable importance only when the velocity exceeds 0.01 ft per hr.

ACKNOWLEDGMENT

The authors wish to thank the several heat pump engineers who have criticized this paper in manuscript form; these include M. S. Oldacre and H. A. Adler of the Utilities Research Commission, Chicago; O. O. Wagley of the Wisconsin Electric Power Co., Milwaukee; and D. M. Vestal, Jr., of the Texas Agricultural and Mechanical College, Reverse Cycle Heating Project. We also wish to acknowledge our indebtedness to Prof. K. J. Arnold and the Computing Service of the University of Wisconsin for the considerable amount of work in evaluating Tables 1 and 2.

SYMBOLS

α = thermal diffusivity = $k/\rho c$, square feet per hour.
$\beta, \gamma, \varepsilon$ = variables of integration.
ρ = density, pounds per cubic foot.
c = specific heat, Btu per (pound) (Fahrenheit degree).
e = Naperian base of logarithms.
$I(x)$ = line source integral. (See Equation 9).
k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
L = latent heat of freezing of soil, Btu per cubic foot.
L_p = length of (short) pipe, feet.
q = rate of heat flow, transfer, or absorption, Btu per hour.
Q' = rate of heat transfer per hour per linear foot of pipe, Btu per (hour) (linear foot).
r = distance from center line of pipe or cavity, feet.
R = radius of pipe or cavity, feet. In the discussion of ice formation, R is the radius of the cylinder of frozen soil around the pipe.
t = time since start of operations, hours.
T = temperature at any point, Fahrenheit.
ΔT = *temperature differential* or difference between the temperature of *surface* of exchanger (or other point under consideration) and *initial* soil temperature, Fahrenheit degrees.
w = rate of heat flow or transfer, per unit area, Btu per (hour) (square foot).

J_0, J_1, Y_0, and Y_1 of Equations 8 and 11 are Bessel functions.

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DISCUSSION

G. B. PRIESTER, Baltimore, Md.: The authors are to be complimented on offering such a fine paper to our industry and profession. Their contribution should be particularly appreciated, since they are not members of our Society.

I wish to state that my associates in the Consolidated Gas, Electric Light, and Power Company of Baltimore, particularly Drs. Guernsey, Betz and Skau, have found the material in the authors' papers useful and helpful. I want to concur with Professor Ingersoll's suggestion that anyone doing work on using the earth as a heat source should carefully review the publications that are available.

MERL BAKER, Lexington, Ty.: From the outstanding work of the authors, I think we are aware of the fact that the ground is an excellent heat source. I would like to have the author comment on the danger of separation between buried pipes and the surrounding soil. Theory indicates that this would be very detrimental. I don't know what effect ice formation may have; however, as water freezes and expands it might cause separation.

W. M. WALLACE, II, Durham, N. C.: I would like to ask Mr. Ingersoll if he has had any experience using finned type pipe buried in the ground.

L. F. DEMING*: In the economic analysis of the value of the heat pump, has consideration been given to the economic life of this underground pipe? How can it be protected from corrosion?

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* U. S. Navy, Bureau of Yards and Docks.

AUTHORS' CLOSURE (L. R. Ingersoll): I should like to thank the discussers for their comments.

Concerning separation, the theory does not say very much about it. I have heard of a case in which a helix was used resulting in a bad separation that occurred, curiously enough, in the summertime. That is the only case I have heard of with a serious separation. The helix is a rather bad form, and has nothing in theory to recommend it.

I do know of a number of cases in which a long, relatively isolated pipe with the hairpin loop was found to work better in the second and third year than it did in the first, and that certainly would indicate that there is no permanent separation.

In reply to Mr. Wallace, two years ago I felt that ordinary pipe was best. Now I am not so sure because recent calculations showed the considerable advantage that you have from pipe *ends*. The end of a pipe gathers heat from the whole hemisphere of the earth behind the end of the pipe and, therefore, it doesn't have such large temperature differentials as the pipe itself. That indicates to me that short, stubby laterals would have their advantage. I think by the same token that fins might also be a good scheme.

To answer Mr. Deming's questions, I don't think any heat pump has been in use long enough so that the exchanger has actually shown serious deterioration. I would expect great deterioration where electrolysis is possible. However, copper pipe is used in plumbing for thirty or forty years. It would appear that it should be satisfactory for exchangers.

J. M. VAN NIEUKERKEN, New York, N. Y.: The only thing I can contribute to this matter of copper deterioration, is that copper tube has been used to a very large extent to replace old and badly corroded iron or steel gas service lines, connecting street mains and the inside piping of homes and other buildings. In numerous instances soft copper tube has been drawn or hard temper tube pushed through the existing service pipes being replaced. Many thousands of copper gas service lines have been in satisfactory operation since the first one was installed about 23 years ago, without any report of trouble, even from locations where, on account of the high sulphur content of the gas, some trouble had been anticipated.

Because of the economy and ease of installing these copper tubes, copper tube is today being used also for new service lines and small gas mains in many cities.

Except in locations where soil conditions are very unsatisfactory, no fear appears to be warranted that the copper tube coils laid in the ground for heat exchange purposes will be the limiting factor in the life of a heat-pump installation.

1416

No. 1416

FACTORS USEFUL IN GROUND GRID DESIGN FOR HEAT PUMPS

By G. S. SMITH*, SEATTLE, WASH.

THE HEAT source or sink often presents a major problem in the use of heat pumps. Where neither air nor water can be utilized, a grid of tubing buried in the earth has often been found satisfactory. On the other hand such grids have often partially or completely failed because of improper size or design. This is not surprising since the ground grid presents a new field of heat exchange design and little is known of the factors involved.

Several experimental studies of this problem have already been made or are now in progress. Perhaps no one study can result in information which can be generally used, since each will be influenced by the climate, locality, type of soil, methods, or apparatus used. However, an accumulation of such information will indicate trends and produce averages sufficiently accurate for most design purposes. Workers in this field of research should be continually aware of the need to present their results in such a manner as to be adapted by designers in any locality.

At the outset, the intent of the present study was to determine, *first,* soil thermal conductivity, *second,* undisturbed earth temperatures, and, *third,* heat-collecting coefficients for buried tubing. As the work progressed, many other factors became apparent and were included. The results indicate that the ground area necessary for any given house in a given climate, is the most important factor, whereas, the heat-collecting coefficients are important mainly in determining the minimum length of tubing needed to collect the heat.

The proper design of a ground grid requires information on a number of items. Those probably most important may be listed in the following order:

First, a reasonably accurate knowledge of the maximum thermal units per unit time necessary for heating or cooling of the structure to be conditioned. This can be quite readily calculated from the house plans and Weather Bureau data, using the standard methods proposed by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Second, a knowledge of the soil characteristics. This includes type of soil, its natural density, the moisture conditions at various seasons, and covering such as grass, etc. Any subsurface moisture migration will generally add greatly to the effectiveness of ground grids unless too many neighboring systems are in use. The major portion of the experimental work and results on soil characteristics has already been presented

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* Professor of Electrical Engineering, University of Washington.

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

in a paper recently published.[1] Later work still in progress has contributed little besides confirming the published results.

Third, reliable information on the heat transfer coefficients for buried tubing at various spacings, time of continued operation, etc., plus some knowledge of the effect of the surface films, and moisture migration due to temperature differences.

Fourth, the average number of hours of actual operation during the heating or cooling season. Such information will aid in selecting proper transfer coefficients, as well as in determining the total heat consumption for the year.

Fifth, annual heat absorption per square foot of soil area, resulting from the sun's radiation, plus that carried in by the rainfall. A small amount of heat for ground

FIG. 1. PARTIAL VIEW OF TEST EQUIPMENT

coils may be supplied by the earth's interior heat, though the distance of travel is so great that such an increment should probably not be relied upon except as a safety factor under extreme conditions.

This report will attempt to present information or methods of obtaining information on most of these items except the *first* and *second.*

DESCRIPTION OF METHODS AND APPARATUS

The determination of the heat transfer coefficients was approached primarily by theoretical calculations and checked experimentally.

Some workers have used the method of images as a means of calculation. This assumes the earth's surface as a constant temperature plane half way between the tube and its reflection at an equal distance above the surface. The curves proposed in this paper are based upon Fourier's heating equation which

[1] Exponent numerals refer to Bibliography.

assumes all points at an infinite distance from the tube to remain at a constant temperature. Neither method represents an accurate solution of the actual case, and both present results more nearly accurate as the depth of the tube below the earth's surface is increased within the limits used in heat pump work.

The equipment used for this study consisted of a 2-hp compressor with the necessary equipment to supply a buried grid of tubing with F-12 gas at any head or suction pressures desired within the range utilized by heat pumps. Fig. 1 shows a partial view of this equipment. The quantity of F-12 flow was measured by means of a calibrated rotometer used on the high pressure side where the refrigerant is still in a liquid state. The building housing this equipment was

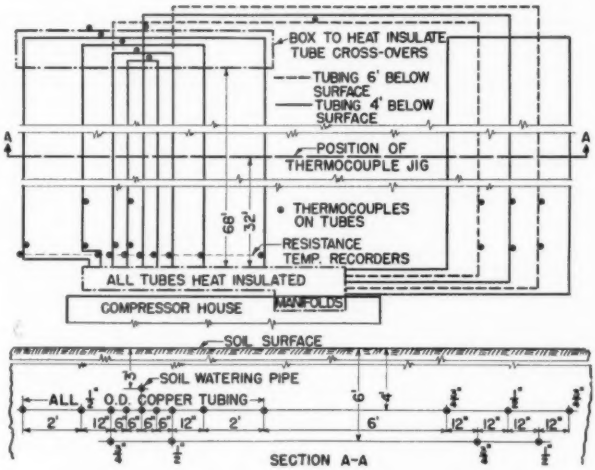

FIG. 2. LAYOUT OF TUBING IN GROUND GRID

well heat insulated and provided with a ventilating fan, as well as an electric heater, each controlled by a thermostat so that the room temperature could be controlled at any desirable point within a normal range and kept relatively constant. The temperatures of the tubes as they entered the soil, as well as those of the outside air, room air, etc., were recorded continually during tests by means of a temperature recorder, with resistance-type thermometers.

Fig. 2 presents the plan and cross section of the tubing, indicating the positions of the various loops, their lengths and depths. Each loop has a cutoff valve at either end, and the group is fed by means of a header carefully designed to equalize the flow for all loops. The loops were so placed that parallel tubes at a variety of spacings could be selected for group operation. The outgoing and incoming ends, as well as the field crossovers, were well insulated with mineral wool against heat transmission to the point where the ground measurements were to begin. One-half-inch outside diameter copper tubing was used for all of the loops except two, and these were provided with ¾-in. tubing. The major

group was placed at a depth of 4 ft; however, two of the loops were placed at a depth of 6 ft.

An unusually elaborate system of copper constantan thermocouples were located in a vertical plane, the position of which is indicated in Fig. 2. A vertical section showing the tube positions and also the various thermocouple positions is shown in Fig. 3. The thermocouples were located very accurately, and held in the proper position with respect to the tubes by means of light wood strip jigs. Thermocouples were also soldered to the various tubes at intervals along their length in such a way that the point of superheating of the gas in the tube could be determined. All couples and leads were well protected with a special synthetic rubber paint and finally by a coating of glyptol varnish.

Temperatures were determined by means of a potentiometer located in a small temperature room, which was held at a constant temperature by electric heat.

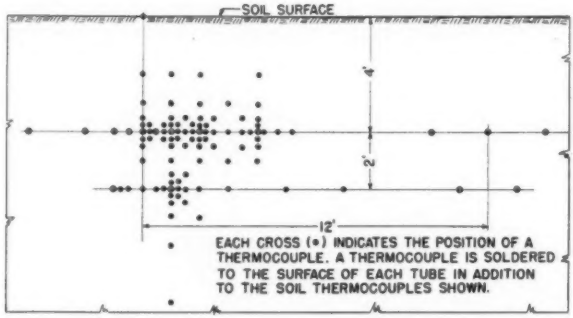
SOIL SURFACE
4'
2'
12'
EACH CROSS (•) INDICATES THE POSITION OF A THERMOCOUPLE. A THERMOCOUPLE IS SOLDERED TO THE SURFACE OF EACH TUBE IN ADDITION TO THE SOIL THERMOCOUPLES SHOWN.

FIG. 3. THERMO-COUPLE LAYOUT

Soil moisture was determined by periodic sampling, though later a moisture cell of established calibration was used.

To determine the normal undisturbed earth temperatures at various depths and seasons, a station with 6 thermocouples at depths of from 1 to 10 ft was located about 50 ft from the test grid. Another such station was located in the back yard on the south side of the author's home about two miles from the test site. A third master station was located more than a quarter-mile away, and consisted of 12 resistance-type thermometers spaced from 1 centimeter to 15 meters below the surface. These temperatures together with air temperatures were recorded continuously.

The length of this report will not permit a complete report on the data collected from these temperature stations, and reference will be made to only certain portions of the results. A report on another phase of this study is in progress which will elaborate these data.

HEAT TRANSFER COEFFICIENTS

For the actual design of a ground grid the rate of heat flow from soil to tube or the reverse, for any given set of conditions, is probably the information most

useful. This interchange of heat is a function of the tube diameter, tube surface conditions, the soil and refrigerant temperatures, as well as soil characteristics.

Ingersoll and Plass[2] have applied the solution of Fourier's differential equation for the flow of heat to or from an infinitely long straight line source. This theory appeared most logical for use in our study. Thus, by using the Equation 1,

$$\frac{U}{k} = \frac{2\pi\Delta T}{I\,(r/2\sqrt{\alpha t})} \qquad \ldots\ldots\ldots\ldots\ldots (1)^*$$

the curves in Fig. 4 are obtained. Although these apply only in the case of a single infinite tube, they are basic in such a study.

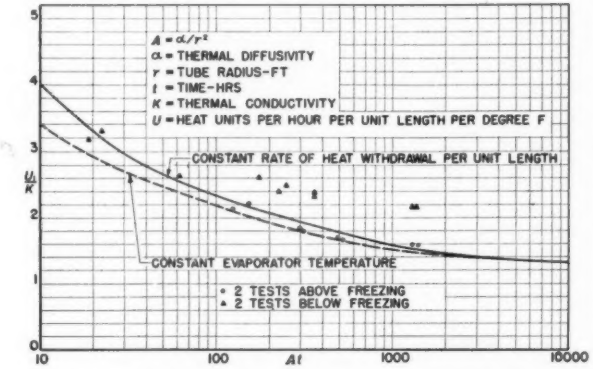

FIG. 4. HEAT TRANSFER COEFFICIENTS—SINGLE ISOLATED TUBE; FULL AND BROKEN LINES CURVES FROM CALCULATIONS; POINTS INDICATED ARE FROM TESTS

The broken line curve indicates the relation of U/k and At, assuming the tube is kept at a constant temperature, while the unbroken line curve gives the similar relation when the heat withdrawal per unit length of tube is assumed constant. While neither curve accurately represents the field conditions, the line curve approaches the actual case very closely.

The coordinates U/k and At may at first appear confusing; however, a little study will reveal the logic of their use. Such plots can be used for any case where the conductivity and diffusivity of the soil is known, and after a given size of tubing has been selected. Curves plotted as U against t can be made only for a given type of soil and size of tube. With known soil characteristics and a given size of tubing, U and t are easily determined by multiplying each ordinate by k and each abscissa by r^2/α. In most cases k will vary between 0.4 and unity or more. For a ½ in. tube and $\alpha = 0.02$ the ratio of $r^2/\alpha = 0.0217$. For a one-inch tube the ratio would be four times as great.

The test points indicated in Fig. 4 are from experimental runs on a single loop with the return tube at such a distance that the tube can be considered an

* See list of symbols.

isolated tube. The points following the curve, and in most cases below, were taken with a tube temperature a little above freezing. The points ranging well above the curve were taken with the tube temperature somewhat below freezing and indicate the increased withdrawal of heat due to the added *heat of fusion* while freezing the soil moisture. Once this heat of fusion is absorbed the points would again fall back to, or near the curve.

SYMBOLS

A = area, square feet.

$a = \dfrac{I(\delta) - I(\beta\delta)}{I(\delta) - 2I(\beta\delta) + I(2\beta\delta)}$

α = thermal diffusivity = $k/c\rho$ square feet per hour.

β = spacing ratio Z/r.

c = specific heat, Btu per (pound) (Fahrenheit degree).

$\delta = \dfrac{\sqrt{\frac{\alpha t}{r^2}}}{2}$

ε = Naperian base of logarithms.

$I(x) = \int_x^\infty \beta^{-1}\, \varepsilon^{-\beta^2}\, d\beta$

k = thermal conductivity; Btu per (hour) (square foot) (Fahrenheit degree per foot).

n = number of tubes.

P = duration of the sinusoidal temperature variation, hours.

Q_1 = flow of heat per unit length of tube, Btu per foot.

Q_2 = flow of heat per unit surface—Btu per square foot.

r = radius of tube in feet.

T = temperature Fahrenheit degrees.

T_0 = maximum value of the annual sinusoidal temperature change.

ΔT = temperature differential, Fahrenheit degrees.

t = time in hours.

U = rate of heat transfer, Btu per (hour) (lineal foot).

U_1, U_2, U_∞ = rate of transfer per number of tubes indicated by subscript.

ρ = density of soil in pounds per cubic foot.

Z = distance between tube centers, feet.

Most check tests were carried on continuously for twenty-four hours or more. The value of k for these check points was in each case determined by the change in temperature per unit distance away from the tube surface by means of the following:

$$k = \frac{Q_1 \log (r_2/r_1)}{2\pi (T_2 - T_1)} \qquad (2)$$

The values of k thus determined are somewhat lower than those determined by the thermal conductivity tests since those calculated by Equation 2 account for a film coefficient in addition to the true conductivity.

The heat-flow equation may be modified, by means of the method of super-

position, for use when two, three or more tubes are used in parallel at any given spacing ratio β. For three tubes in parallel the equation is:

$$\frac{U_3}{k} = \frac{2\pi}{2a\, I(\delta\beta) + I(\delta)} \times \frac{2a+1}{3} \quad \ldots\ldots\ldots (3)$$

and for an infinite number of tubes:

$$\frac{U_\infty}{k} = \frac{2\pi}{I(\delta) + 2\sum_{n=1}^{n=\infty} I(n\beta\delta)} \quad \ldots\ldots\ldots (4)$$

Fig. 5 shows the theoretical curves calculated for three tubes operating in parallel at spacing ratios of $\beta = 24$ and 48. The points indicated are those found by experiment.

In general a grid cannot be constructed so as to be considered a single tube and in most cases would be greater than two or three parallel tubes. Rather than to attempt to derive a formula for the desired number of parallel tubes, two sets of curves were prepared. These are presented in Fig. 6, giving the ratios of heat transfer of three parallel tubes to that of a single tube in line curves, and the same ratios for an infinite grid to a single tube in broken-line curves. If the tube-spacing ratio is reasonably large and the time factor not too great the amount of heat extracted by a unit length of tube for a given set of conditions can be interpolated as accurately as other necessary factors can be determined.

TUBE SURFACE CONDITIONS AND MOISTURE MIGRATION

During the studies to obtain heat transfer coefficients as well as those on soil thermal conductivity, the temperature gradients in the soil near the metal surfaces were found to be abnormal as compared to those in the main soil volume. This non-linear temperature gradient will decrease, to some extent, the heat flow into the tube. In order to study this effect in more detail, a special soil container 7 in. in diameter and 21 in. long was used, with the covers or ends made of insulating board. A copper tube of the desired diameter and full length of the container was placed at the center of the container, and a second tube, ¼ in. in diameter, within this. A special fitting was made to carry the cooling liquid up through the ½-in. tube and let it return through the larger tube. Thermocouples were arranged to measure carefully the ingoing and outgoing liquid temperatures, and the liquid quantity was measured by timing and weighing the flow. Within the soil body a light plastic strip held thermocouples at definite distances between the tube and container walls with spacings as small as 1/16 in. near the metal surfaces. Thermocouples were also soldered to the tube and container surfaces.

Fig. 7 shows plots of the various temperatures at their corresponding distance from the center, when a tube of ½-in. in diameter was used. After 9½ hours running, conditions were very nearly constant. On the cold side the film effect extended to less than 1/16 in., while on the outer or warm side a somewhat greater distance was involved. When the soil adjacent to the tube was not packed carefully the effect often extended to ⅛ in. or more.

The apparatus was not constructed carefully enough to permit accurate determination of thermal conductivity, but by using the k found for this soil by other methods, the relation of the true k to the apparent k_a can readily be determined. This depends upon the slope of the line drawn from the tube temperature, to the point on the true soil gradient line at the reference distance chosen. For a 2-ft spacing of tubes this distance would be 1 ft from the tube center. From these tests as well as from observations on the heat-transfer tests, the apparent conductivity up to a one-foot radius dropped on an average of 5 percent to 10 percent below the true conductivity. For a lesser radius the difference would be greater and for a greater radius the difference would be less. Time has not allowed an investigation as complete as can be desired, but from the various

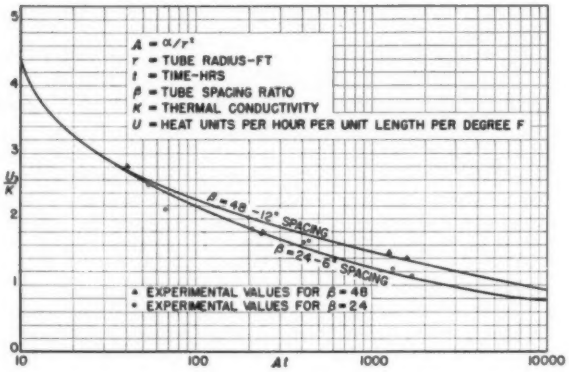

FIG. 5. HEAT TRANSFER COEFFICIENTS FOR THREE PARALLEL TUBES; CURVES ARE FROM CALCULATIONS; POINTS INDICATED ARE FROM TESTS

tests made, the same general conditions seem to prevail rather closely for various soils, and tube sizes. A larger heat withdrawal appeared to increase the effect slightly.

No direct effort was made to determine moisture migration. However, its effect could be detected in the various laboratory tests and probably accounts for the larger surface effect at the warm surface. In ground grid work moisture migration will generally aid in heat absorption for space heating, but would reverse its effect for space cooling.

OTHER FACTORS AFFECTING GRID OPERATION

The major experimental work was made at a depth of 4 ft. A preliminary installation at 2 ft was first used, and a few runs were carried out at 6 ft. So long as the soil is initially at a uniform condition, and the run is not long enough to absorb the heat at any relatively great distance from the tube, the results seem to check quite closely. In general a tube near the surface will be more quickly

affected by the atmospheric conditions above the surface and would be most desirable in a temperate climate where the sun's radiation is available during a considerable portion of the heating season. For colder climates and regions where the sky is overcast most of the heating season, a greater depth is desirable. Probably only in climates where the winter is severe and the summer very bright and warm would depths below 6 or 8 ft be desirable.

The effect of surface atmospheric conditions is clearly indicated in Figs. 8 and 9. These are plots of constant temperature surfaces about the tube. The lines originating at the tube surface which are perpendicular to each of the constant temperature curves, indicate the direction of the flow of heat. By spacing these

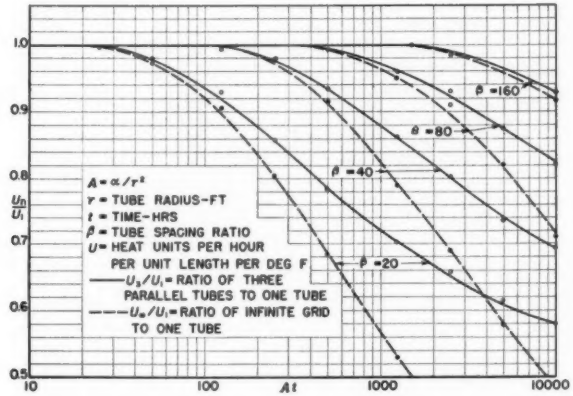

FIG. 6. HEAT TRANSFER COEFFICIENT RATIOS; CURVES AND POINTS ARE FROM CALCULATIONS

properly each represents an equal increment of the total heat flow. Fig. 8 was taken after a 24-hour run during an exceptionally cold spell in January, while Fig. 9 was taken after a similar run when the ground was receiving a plentiful supply of heat from the spring sunshine. A series of such plots were made and offer an interesting means of studying the operating conditions as affected by the time of continuous operation by parallel tubes at various spacings, and by the atmospheric conditions above the soil surface.

A perforated tube shown in Fig. 2 was installed about 6 in. above the middle portion of the grid at a four-foot depth. During the dry summer season, about 200 gal of water was dissipated through this tube after 52 hours of continuous run when fairly constant conditions had been attained. Within 40 hours the flow of heat had increased about 35 percent and was still about 12 percent high after 286 hours continuous running. This change is largely caused by increasing the soil conductivity, but could be partially the result of heat added by water warmer than the soil. Rain during the heating season, or moisture during the cooling season is almost always advantageous.

DETERMINATION OF OPERATING TIME

The designed size of a heat pump or other heating unit must be based on the continuous heat necessary for the assumed coldest day. Operating at this output, the unit will be in actual use less and less of the time as the mild weather approaches. Hence, a knowledge of the operating time becomes very desirable. This can be determined with moderate accuracy by using Weather Bureau information. Thus Table 1 gives from 50 years of records, the average degree days for each month of the year at Seattle, Wash. The degree days for the coldest

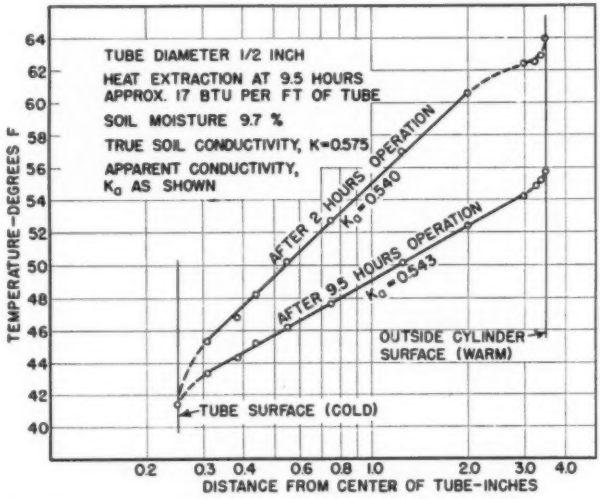

FIG. 7. CURVES SHOWING NON-LINEAR TEMPERATURES AT SURFACES

day in January for each of three unusually cold Januaries, were averaged, as the average coldest day to determine a degree day factor which would be proportional to the size of the heating unit needed. Thirty days multiplied by this average coldest day would be the equivalent monthly degree days, assuming each day equally cold. The actual average degree days for any month divided by this heating base value, times 100, will give the percentage time of operation for the heating unit.

From records of four houses in Seattle using heat pumps, the percentage of operating time was calculated and is indicated in Table 1. In two of these houses the unit was either somewhat larger than usual, or the occupants were very conservative with heat. House D appears to have too small a unit as indicated by the high percentage of operating time. In general the calculated values check reasonably well with actual operation.

HEAT AVAILABLE FOR GROUND GRIDS

The success of a ground grid depends as much on the heat available in the volume of soil supplying that heat, as it does upon the tube size, spacing, length, depth, etc., or even more. The available heat must be supplied by the heat radiated from the sun, the heat carried into the soil by the rain, underground moisture migration and heat from the earth's center. If the heat absorbed each heating season is not, on the average, replenished each year the unit will, in time, fail.

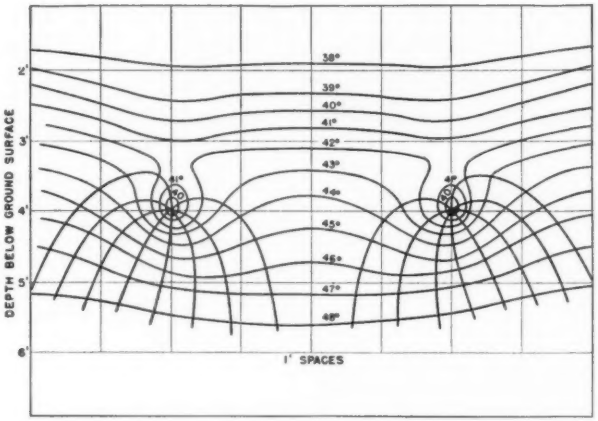

FIG. 8. ISOTHERMAL LINES AND HEAT FLOW PATHS FOR JANUARY; PARALLEL TUBES SPACED 4 FT APART

Fig. 10 shows a plot of the temperatures near the tube at various periods before and during the operation of the experimental unit. Since these were taken in July they are somewhat affected by the incoming heat from the sun. During the colder portion of the year they would show temperature changes to somewhat greater distances. Fig. 10, as well as plots like Figs. 4, 5, 8 and 9, indicates that in general the heat does not travel very great distances except under extremely long operating times, and then only very slowly. This would largely eliminate the heat from the earth's center since the temperature at 30 to 50 ft remains at about the average of the yearly temperature above the surface, and the temperature at greater depth changes only about one degree for every 60 ft. This great distance and small temperature differential will result in very little heat migration.

The underground moisture migration might or might not be present. If other ground grids were near, it might be of little value or even a disadvantage. This leaves only heat from the sun and rain. To determine how much of this heat can

be utilized by the ground grid would require rather extensive studies and even then could be only approximated.

From the considerable amount of information available from the temperature stations described, some rather interesting studies were made, which have resulted in some possible guiding limits.

Again using Fourier's differential equation and the cyclic temperature variations during the year, the following formula for calculating the possible heat

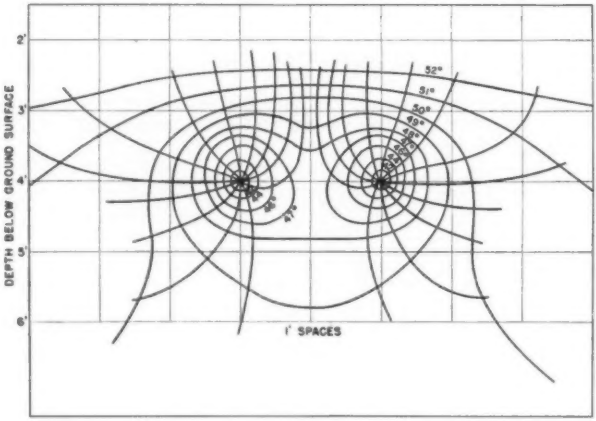

FIG. 9. ISOTHERMAL LINES AND HEAT FLOW PATHS FOR MAY; PARALLEL TUBES SPACED 2 FT APART

absorbed by each square unit of soil during the summer months can be derived:

$$\frac{Q_2}{A} = k T_o \sqrt{\frac{2P}{\pi\alpha}} \quad \ldots\ldots\ldots\ldots\ldots (5)$$

Where no heat is extracted from the earth, the temperature differential causing heat to flow into the soil must be based on the constant temperature at 30 to 50 ft below the earth's surface. Where a ground grid has extracted the heat nearer the surface the soil temperature will be at a much lower value.

Fig. 11 shows a plot of soil-solar-heat absorption calculated by means of Equation 5 using a maximum temperature, T_o, obtained by finding the equivalent sinusoidal temperature difference between the soil surface and the assumed average soil temperature. These calculations were carried out using the recorded temperatures at the surface of the soil for two summer seasons. Similar calculations were found to give approximately the same results by using 71 percent of the Weather Bureau's maximum daily temperatures instead of the actual surface temperatures, as an approximate method of analysis, and points are plotted to indicate these approximations for the same two years. Fig. 11 also presents the possible heat added due to rainfall. This curve was calculated by

using the Weather Bureau's 50 years monthly average rainfall together with its corresponding monthly average air temperature, and assuming the moisture precipitated has the same average temperature as the air. The procedure in obtaining the curves in Fig. 11 will be fully presented in the paper previously mentioned.

There is no intention to intimate that the suggested method of arriving at the available stored heat is accurate and of course the figures presented apply

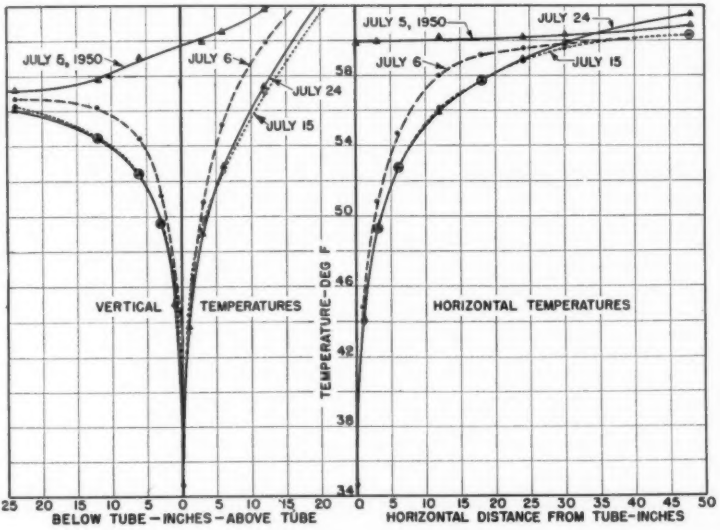

FIG. 10. TEMPERATURE GRADIENTS NEAR TUBING AFTER VARIOUS PERIODS OF CONTINUOUS OPERATION

only to the test site. In sections of the country where summer rains are frequent and the surface temperatures due to solar radiation are higher a somewhat greater heat penetration will result.

In most temperate climates any rains during the heating season probably add a considerable amount of heat that has not been accounted for in the calculations. At any time when rainfall can occur, the temperature will usually be above freezing, and with the soil somewhat below normal temperatures because of heat extraction, nearly all available heat in such precipitation will be absorbed by the soil. At the same time the average monthly temperature might appear to be so low that little or no useful addition of heat would be indicated by the method proposed.

A similar argument would apply during the winter months where bright sunshine might radiate a sizable amount of heat to the soil even when the air temperatures are rather low. Such additions might well be approximated by a

careful study of U.S. Weather Bureau records which present rainfall, and sunshine conditions daily.

Until experience in each locality can determine the approximate amount of heat that can be extracted annually per square foot of surface, the method proposed, or some similar method of analysis might be used to avoid failures.

SUGGESTIONS FOR APPLICATION OF DATA

The design of a ground grid at this time still requires a number of assumptions which must be based upon the best information available. This study seems

TABLE 1—HEAT PUMP OPERATING TIME IN PERCENTAGES FOR SEATTLE
(Based on Averages from 1900-1950 Weather Bureau Records at Seattle, Wash.)

MONTH	DEGREE DAYS	PERCENT OF TIME OPERATING[a]	HOURS OPERATING	PERCENT OF TIME OPERATING			
				HOUSE			
				A	B	C	D
Jan.	763	60	446	29	35	55	93
Feb.	635	50	360	31	54	50	69
Mar.	590	46	324	22	35	33	61
Apr.	434	34	224	19	22	31	
May	289	22	163	10	9	9	
June	157	12	86	3	7	1	
July	65	5	37	0.6	3	1	
Aug.	68	5	37	1	3	1	
Sept.	169	13	93	3	5	8	
Oct.	365	29	215	11	20		
Nov.	556	43	301	22	29	43	42
Dec.	704	55	409	27	29	61	81
Total	4795		2695				

[a] Percent based on continuous operation of units for the average coldest day in the three coldest Januaries during the last 50 years, not including 1950.
Maximum January degree days = 41.3 × 30 = 1239.

to indicate that the safest method of determining the adequacy of a given plot to supply a given heat load, is to calculate as nearly as possible the amount of heat it will probably absorb during the summer season and depend upon this stored heat for the major portion of the heating season. In most climates some heat due to the sun's radiation, as well as precipitation, will be available during most of the year, and especially in early autumn and later spring, even while heating is still required.

To illustrate some of the possibilities, assume a 10,000-Btu per hour increment as a basic heating unit for a home in Seattle. The major heating period might be considered from October 1 to April 1. Before and after those dates the additional absorption of heat due to the presence of a ground grid would probably supply the heat necessary. Between those two dates only the stored heat would be assumed available, though actually some useful heat may still be added by the sun's radiation or rainfall. From Table 1 this would require a total heating time, October 1 to April 1, of 2055 hours or 20,550,000 Btu of heat. Assume the average temperature of the body of soil involved will drop to 40 F. Fig. 11 shows that the stored heat due to the sun's radiation would be 9,500, and that

due to precipitation, 1500 Btu or a total of 11,000 Btu per square foot of soil surface. The area required for this heating unit would then be slightly over 1875 sq ft or a strip about 19 by 100 ft.

Next a grid for this plot must be designed of such tube length and spacing that it will absorb the desired amount of heat per hour under the most adverse conditions in January. The month of January will require a total of 446 operating hours, or operation about 60 percent of the time. Experience from these

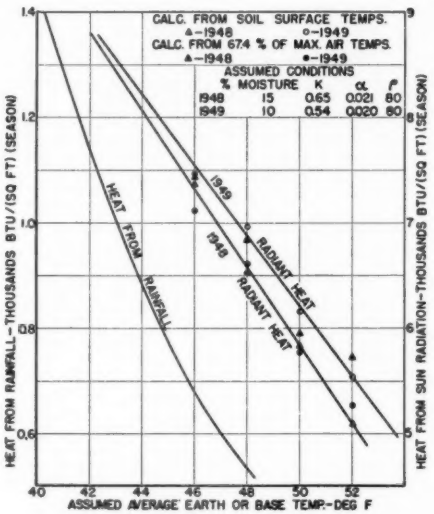

FIG. 11. CURVES OF SOIL HEAT ABSORPTION FROM SOLAR RADIATION AND RAINFALL TABLES

tests as well as reports from other tests indicate that it requires two to four times the actual continuous running time for the soil body to recover, or very nearly recover, uniform temperature conditions. Just how far down the transient curve of Fig. 4 the usual intermittent operation would carry would be rather difficult to determine accurately. However, to assume that this point is well toward the portion where the curve approaches the horizontal would appear quite safe. This is assumed to be at $At = 5{,}000$ and $U_1/k = 1.4$.

For the soil conditions during the heating season, assume a sandy soil with an average moisture content of 15 percent, a dry density of 93 lb per cubic foot. From the thermal conductivity tests, k would be about 0.95 and α would be 0.035 if the specific heat of the dry soil is assumed as 0.19. The specific heat for the moist soil would then be $0.34/1.15 = 0.295$. The film or surface conditions near the tube will have the effect of decreasing the apparent thermal conductivity, though moisture migration due to the cooled tube surface would

tend to counteract such a decrease to some extent. Since there is no known method of determining the accurate extent of these effects, assume the conductivity has been lowered to 0.85, or about a 10 percent change. This is about the magnitude of change found in the experimental work described.

The heat absorption factor U_1 for single tube operation would then be the product of k and U_1/k or $0.85 \times 1.4 = 1.19$ Btu per (hour) (foot length) (degree Fahrenheit). However, to draw upon the total area, parallel tubes must be used and the absorption factor will be still further reduced. As a rough guess, assume a 2-ft spacing would be possible, and that ½-in. tubing is to be used. This would give a spacing ratio β of $24/0.25 = 96$. The heat-transfer ratio curves in Fig. 6 show the ratio at $At = 5000$, and $\beta = 96$ would fall near 0.9 as a mean between that for three tubes and for an infinite grid. Using 0.9 of the single tube value 1.19 gives a final design value of 1.1.

For the highest coefficient of performance, the temperature of the grid should be maintained as high as will produce the necessary heat. In theory this evaporator temperature should be lowered as the average ground temperature drops. Difficulties in automatically producing such adjustments will result in either

TABLE 2—DETAILS ON DATA PRESENTED IN TABLE 1

HOUSE	GROSS HEATED CU FT	UNIT HP	HEAT SOURCE	YEAR	DEGREE DAYS, JAN.	DEGREE DAYS MAX. DAY
A	10904	3	Well	1916	1056	43
B	39000	2.5	Grid	1936	1000	41
C	13000	3	Lake	1937	1032	40
D	33500	5	Spring		Avg.	41.3

periodic manual changes or a permanent setting for the critical heating period. A small amount of freezing about the tube will result in no harm and will provide a greater heat pickup per length of tubing as well as some additional heat from the *heat of fusion* of the soil moisture. Let us assume the grid surface temperature is dropped to 20 F, which for F-12 would require an evaporator pressure of about 21.1 psi. With the previous assumption that the average soil temperature would be about 40 F, the temperature differential would then be 20 deg. This would allow a minimum heat withdrawal of 20 times 1.1 or 22 Btu per hour per foot. This would require 455 ft of tubing. Five parallel tubes in the 20-ft width would allow a 4-ft spacing and this would in turn increase the heat-transfer ratio factor to near that for a single tube. Thus 450 ft would appear about correct. While this grid may prove to be a little over designed, such fault is to be desired until enough practical operating experience has been accumulated to afford more accurate designs.

In general the tubing should not be placed beneath buildings or other close coverings since such soil volumes must receive heat replacements from a horizontal direction and only at the expense of adjacent plots. Severe freezing under or near buildings has resulted in considerable damage due to heaving of soil. Trees and thick vegetation coverings will also tend to reduce the annual heat absorbed by the soil. On the other hand high moisture content and subsurface moisture migration will as a rule greatly add to the available heat.

CONCLUSIONS

While accurate information for ground grid design is very difficult to obtain, and many assumptions must be made, the results definitely indicate that the average city lot is too small to support the house usually built thereon. Thus, the ground grid is practicable only where larger lots are available, or where very special soil, moisture and subsurface conditions are found. Soil densities, conductivities, etc., should be carefully chosen since the soil around a newly installed grid may require years to attain its original density. The use of vertical tubes in holes dug quite deep can be used if the spacing and number are such that a sufficient volume is uniformly drawn upon. While the greater depths may draw on greater volumes, if the heat is not returned each season the design is in danger of ultimate failure unless moisture migration at these greater depths assists by drawing from neighboring areas.

Every effort should be made to add to the present data by careful studies of both successes as well as failures where ground grids have been in actual use.

ACKNOWLEDGMENTS

The investigation reported in this paper as well as the previously reported work on soil thermal conductivity was made possible by grants from the Bonneville Power Administration, and the Engineering Experiment Station at the University of Washington. The author wishes to acknowledge the valuable assistance of L. C. Sutherland, J. H. Whiting, R. O. Berglund, and other graduate students who carried out much of the experimental work and mathematical calculations.

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DISCUSSION

R. C. CHEWNING, Portland, Ore. (WRITTEN): In making this criticism of Professor Smith's paper, I would like to make it clear that all comments are based on observations of actual heat pump installations, and, while explanations of some phenomena are suggested, this is done without any exact knowledge of how and why heat is transferred from the ground to a pipe used as the exchanger for a heat pump.

Between 1945 and 1949 a total of 112 heat pumps were installed in Portland, Oregon, and surrounding towns. Of these, approximately 60 have been of the ground coil type, 28 of which are still in operation. Some have five years of experience behind them. Opinions and suggestions expressed in this criticism are based on general observation of the operation of these installations, and a somewhat detailed observation and study of one of them selected as the object of a chapter research project.

Of 37 ground source heat pumps installed by one contractor, 29 failed because his ground coil design was based upon the amount of metal surface required to transfer the necessary heat from the ground to the refrigerant with the ground temperature remaining constant. Another contractor, Mr. U. G. Smith, realized that the storage and heat regain capacity of the ground was limited and spread his coils under a much larger area. In this case, 20 of his 23 installations are still in operation.

With one exception the calculations and factors presented by Professor Smith agree very well with the experimental data which he obtained. The exception is the experimental data presented in Fig. 4 of his paper, in which the evaporator temperature was below freezing. In this case the experimental data plotted well above the calculated data. Prof. Smith states that the discrepancy is due to heat of fusion withdrawn from the soil under these conditions, and that once the heat of fusion is absorbed the experimental points would again approach the calculated curves. Professor Smith presents no experimental data to substantiate this statement.

This particular point is discussed here because there is such a wide variance between the coil requirements indicated by Professor Smith's calculations when compared to the design of actual installations which have accomplished the purpose intended. It is thought that phenomena occuring in the soil surrounding a heat pump ground coil when the soil is frozen may explain the difference in theoretical calculations and actual operation.

According to Prof. Smith's analysis, a home in Seattle with a 10,000 Btu per hour *basic heating unit*, by which I understand a 10,000 Btu per hour design heat loss, would require a coil of 450 ft of ½ in. tubing covering approximately 1900 sq ft of ground. The house observed by the Oregon Chapter's Research committee has a 38,000 Btu per hour design heat loss and has been maintained at 72 to 75 F for three heating seasons by a coil constructed of 1000 ft of ⅞ in. O.D. tubing covering approximately 2000 sq ft of ground. The coil operates at a suction temperature of approximately 20 F during midwinter and freezes the ground for a 6-in. to 12-in. radius around the liquid supply end of each coil in the grid. This amount of freezing in a ground coil has no ill effects if the coil is properly installed. This installation is generally quite similar in design conditions to the other successful heat pump installations made by Mr. U. G. Smith.

Four phenomena which might occur to explain the discrepancies between theory and actual operation are:

1. Freezing of the ground adjacent to the coil causes a considerable reduction in the water vapor pressure at the tube in the soil, which results in a movement of water vapor and the heat contained in it toward the frozen area.

2. Freezing of the soil adjacent to the tube creates compressive forces around the tube, eliminating, at least partially, the otherwise poor transfer at the surface of the tube.

3. Freezing the soil may tend to increase the k factor of the soil.

4. When the soil is frozen, the movement of heat from the surrounding unfrozen area to the frozen area immediately adjacent to the tube continues unabated after the heat pump stops because of the storage effect of the heat of fusion.

Professor Smith's method of determining the area or volume of ground required for a ground coil is probably more misleading than the amount of overdesign which would result from the use of heat transfer factors suggested by him. The fallacy of the theory that a ground coil can supply heat only in relation to the quantity which it can store during the summer is illustrated in the following example. The example also illustrates the relative importance of heat from the sun and heat from rain under conditions of actual operation.

During January, 1949, Portland was subjected to a 16 day cold spell during which less than 0.15 in. of moisture fell, the average daily maximum temperature was approximately 35 F and the average sunshine was approximately 50 percent of the

possible. One of the ground coil heat pumps in Portland started to have trouble. The surface of the ground was not shaded, was solidly frozen, and at no time gave any evidence of thawing. This was followed by a week during which 6 in. of snow fell and remained on the ground. Then came two days with 3 in. of rain and a maximum temperature of 49 F. The coil recovered immediately and finished out the heating season. The coil in this case covered approximately 1000 sq ft and served a residence with a 34,000 Btu per hour design heat loss. The following heating season was even more severe and this heat pump was removed. The significant fact is that, although the ground heat source was exhausted at the end of January, it recovered and finished out the heating season satisfactorily. According to Professor Smith's recommendations, the ground source for this installation should have covered approximately 6400 sq ft. It came very close to being satisfactory with only 1000 sq ft.

Again, a contributing factor in Professor Smith's thinking on the factor of ground heat storage and regain may have been the fact that his experiments have emphasized conditions obtained when evaporator temperatures are above freezing rather than below it.

A. K. EID, Cairo, Egypt: I would like Professor Smith's opinion on the use of pipe placed vertically in the soil. This might be useful in the cities where space is at a premium.

In my work we have had difficulty with oil from the compressors and the problem of getting rid of it. Perhaps the author can explain how he overcomes this problem.

As a final question, how long would the pipes last, based on observations from the author's experiments?

L. R. INGERSOLL, Madison, Wis.*: I am naturally very grateful to Mr. Smith for this experimental working out of certain phases of the exchanger problem in which his work agrees with some of our theory.

I would say, concerning his last remarks on whether the summer heating will compensate for the winter cooling, that I know myself of no case where it has not been completely compensated by the summer heat. I did a little calculating for the Wisconsin Electric Power Company on this matter and my results came very close to their experiments in the matter. There was no trouble about compensation at all, or no need for special summer heating by auxiliary apparatus.

G. B. PRIESTER, Baltimore, Md.: I would like to ask the author how he could be sure that the grid design in his paper meets the requirements when his tests were conducted for such a short period of time.

AUTHOR'S CLOSURE: I have been able to follow, to a limited extent, some of the excellent work Mr. Chewning and his associates have accomplished with their studies of heat pump installations in the Portland, Oregon territory. They are making a fine contribution and should give us all the results of their findings, for it is only by a combination of theory and actual experience that the final solution will be obtained.

Mention was made in the paper that the proposed design was probably pessimistic, and was purposely made so. Mention was also made that the earth probably received a considerable amount of heat from winter rains, or even sunshine, which was not considered since no way was found to predict the amount. Until such additional heat can be predicted, or sufficient information has been accumulated from actual installations, we should base designs upon information which will assure satisfactory operation, and limit ground area only when we are assured less will accomplish results. The amount of tubing necessary may be reduced in a similar manner.

I can agree with Mr. Chewning on some of his suggestions about freezing the soil. There appears to be no disadvantage in freezing the soil for a limited distance about

* Professor of Physics, University of Wisconsin.

the tubes so long as building foundations are not disturbed. The thermal conductivity of the soil is not appreciably altered due to freezing, and Miles Kirsten has found the conductivity increases slightly at temperatures well below freezing. The soil expansion due to freezing should tend to decrease the film effect at the surface of the tube.

However the fact that some very limited areas do prove satisfactory may sometimes be due to factors not always apparent. I have observed two cases in Seattle where unusual results have been obtained, but in both cases the movement of underground water on a hillside slope appears to be the explanation. A layer of hard pan just below the grid could retain an unusual amount of moisture, and thus heat storage around the grid, as well as an increased thermal conductivity from below due to the density of the hard pan layer.

I can see no reason why freezing the soil can give any other advantages than, first, to add the heat of fusion, which is limited to the amount of soil frozen, second, slightly increase the thermal conductivity and decrease the film effect, and third, increase the absorption of heat due to rainfall or winter radiation from sunshine. The Portland area probably has warmer rains and more winter sunshine than the Seattle area since it is further inland from the Ocean.

Mr. Eid suggests vertical pipes rather than horizontal tubes. The vertical pipes might be more economical to install, and would doubtless prove as satisfactory but would require about the same soil surface area since the heat for each season must, on the average, be supplied that same year and only a sufficient area of unobstructed soil surface will accomplish this result.

In our experimental apparatus we used a well designed oil separator just after the compressor and appeared to experience no trouble with oil in the grid. In fact we used a rotometer to measure the liquid freon flow, in which meter the liquid passes through a glass tube. No oil was apparent in this glass tube.

The duration of our tests was only about three years but unless some chemical element in the soil is of such nature as to corrode the tube there appears to be little limit to the time a copper tube will operate. If an anti-freeze solution of water were used in the grid instead of the refrigerant, good grade iron pipe with an outside protective coating would probably last for the life of a compressor.

I can quite readily agree with Professor Ingersoll that unless there is too much obstruction on the soil surface in the form of trees, brush or heavy vegetation, the summer heating will easily replenish the heat removed during the winter. Scientists have calculated that each square foot of the earth's surface receives on an average, about four Btu per minute while the sun is shining directly upon it. This could amount to several hundred thousand Btu per square foot during the summer season. Only a small portion enters the soil and even less reaches any great depth. However, the cooler the ground in the spring the more readily this heat absorption will take place. On the other hand a given volume of soil at a given temperature and moisture content can hold only a definite quantity of heat; any excess heat impinging upon the surface must be absorbed elsewhere.

If I interpret Mr. Priester's question correctly I would say that we ran tests on the grids for a good portion of three years though not attempting to duplicate the operation of an actual installation. The main objective was to substantiate the calculated results and this was apparently accomplished. Most of our tests were for a period of 24 hours though we ran some as long as two to three weeks.

Since the calculated results can be carried to any length of continuous operating time desired, the only remaining problem is to choose a point on the curves which corresponds to the intermittent operation of an actual heat pump. This equivalent continuous operating time is related to the integrated short periods of operating time as compared to the integrated non-operating periods. In the design outlined, this point was taken well down toward the horizontal portion of the curve where an increase in operating time does not greatly alter the heat transfer coefficient.

1417

No. 1417

SOLAR ENERGY TRANSMITTANCE OF FIGURED ROLLED GLASS

By G. V. PARMELEE* AND W. W. AUBELE**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

A CONTINUING program of research to develop design heat flow data for glass in its various architectural forms has been carried on since 1944 at the A.S.H.V.E. Research Laboratory under the direction of the Committee on Research and the Technical Advisory Committee on Heat Flow Through Glass†. Previous papers[1, 2, 3] have given solar energy transmittance data for flat glass and glass block. This paper presents similar data for seven patterns of figured or patterned glass. These data will be used in the preparation of design heat flow tables like those recently published for flat glass and glass block[4].

EXPERIMENTAL PROCEDURE

The Calorimeter: Except for minor changes, the calorimeter used in these studies and the instrumentation were essentially as described in the previous reports. In addition to the usual measurements of solar intensity, measurements

TABLE 1—DESCRIPTION OF FIGURED GLASS PATTERNS

No.		IN
11	Hammered heat absorbing glass	1/4
12	Hammered and etched heat absorbing glass	1/4
15	Ribbed flint glass (shallow ribs)	7/32
16	Ribbed flint glass (deep ribs)	7/32
17	Diffusing flint glass (small circular indentations)	7/32
20	Hammered flint glass	7/32
21	Hammered and etched flint glass	7/32

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† Personnel: R. A. Miller, *Chairman;* A. B. Algren, W. J. Arner, A. H. Baker, F. L. Bishop, Jr., R. D. Blum, J. E. Frazier, J. S. Herbert, E. R. Koppel, D. R. Muir, F. W. Preston, W. C. Randall, C. A. Richardson, Vic Sanders, H. B. Vincent.

[1] Exponent numerals refer to References.

Presented at the 57th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1951.

were made, in most tests, of the low temperature radiant energy received from outdoor surroundings by the glass surface. A specially constructed convection-compensated radiometer was used for this purpose.

The Test Specimens: Single sheets of patterned glass, as described in Table 1, were studied in this investigation. The patterns of samples 11, 12, 17, 20 and 21 were sufficiently random in orientation that the influence of solar azimuth on transmittance at a given solar altitude was considered negligible. Test results

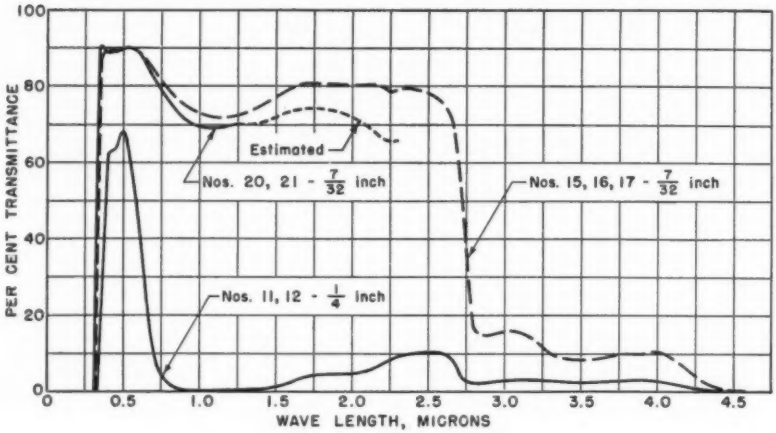

FIG. 1. SPECTRAL TRANSMITTANCE CURVES OF FLAT POLISHED SPECIMENS OF GLASS IDENTICAL IN COMPOSITION WITH FIGURED GLASSES STUDIED

bore this out. On the other hand, samples 15 and 16 were ribbed so that the transmittances of these sheets could not be defined in terms of incident angle alone. These sheets were set in the calorimeter frame with the ribs vertical and tests were run in such a way as to bring the sun's rays parallel, at 45 deg, or at 90 deg to the direction of the ribs. In all cases the patterned surfaces were on the weather side with the rough rolled surface facing the calorimeter shell.

As in the experimental work on glass previously reported, glass surface temperatures were measured with thermocouples cemented into shallow grooves with Canada balsam cement.

RESULTS OBTAINED

Spectral Transmittance Characteristics: Flat polished specimens taken from the same sheets as those studied in the calorimeter were examined to determine their spectral transmittance characteristics; the results are shown in Fig. 1. Values for flint glass have been corrected to 7/32 in. thickness and those for heat absorbing glass have been corrected to ¼ in., since these are the nominal thickness of the samples listed in Table 1. Table 2 compares the normal inci-

dence transmittance values for these specimens computed for Moon's[5] energy distribution for air mass = 2 with transmittance tests made on small flat polished specimens in direct sunlight.

Transmittance of Solar Radiation: Transmittance values for diffuse solar radiation as determined by calorimeter tests are given in Table 3. In addition to giving the range of values determined and the average of the various tests, the table also gives computed values and values recommended for design use. The computed values were determined by integration over a quarter sphere of the product of the energy received at a given incident angle times the transmittance of the glass for direct solar radiation at that angle of incidence. It was assumed that the sky vault was a uniform radiant energy source. The design values are those which gave the least scatter of test points when transmittance of direct radiation was plotted against incident angle.

Transmittance values for direct solar radiation as a function of incident angle are given in Figs. 2 through 6. As noted on the figures, test points for samples

TABLE 2—TRANSMITTANCE VALUES OF FLAT POLISHED SPECIMENS OF FIGURED GLASS FOR NORMALLY INCIDENT DIRECT SOLAR RADIATION

Sample No.	Tests in Sunlight with the Smithsonian Silver Disk Pyrheliometer					Calculated for Standard Energy Distribution[a]	
	As Tested			Corrected Values			
	Thickness In.	Range in Trans %	No. of Tests	Thickness In.	Av Trans %	Thickness In.	Trans %
11,12	0.197	26.1-27.9	2	1/4	19.5	1/4	24.1
15, 16, 17	0.159	79.8-80.7	2	7/32	76.4	7/32	82.0
20, 21	0.190	77.1-80.6	3	7/32	77.1	7/32	80.3[b]

[a] Moon's[5] energy distribution for air mass = 2.
[b] Spectral values > 1.25 μ estimated.

11, 12, 17, 20 and 21 include data from both fixed incident angle tests and fixed orientation tests. The agreement between the test points and the curves indicates that these patterns had no special orientation. Hence, the transmittance curves are functions of incident angle only. The curves for samples 15 and 16 show, however, that the transmittance will vary with different combinations of solar altitude and azimuth angles that give the same incident angle.

Total Heat Gain from Sunlit Glass: In most practical cases in the field of air conditioning, glass is opaque to all radiation except that from the sun or from lighting installations. This is illustrated by the spectral transmittance curves of Fig. 1 which show that the transmittance is practically zero for wavelengths of 4.5 μ (microns) or longer. Even from radiant energy sources at temperatures as high as 450 F only a negligible amount, approximately 2 percent, is transmitted by glass. If lighting sources can be neglected it is convenient to express the heat flow through a glass section by the following heat balance equation in which the heat flow quantities are for a unit area and a unit time:

$$\left[\begin{array}{l}\text{Total heat flow}\\ \text{through glass}\\ \text{section}\end{array}\right] = \left[\begin{array}{l}\text{Transmitted}\\ \text{solar radiation}\end{array}\right] \pm \left[\begin{array}{l}\text{Heat flow by con-}\\ \text{vective and radia-}\\ \text{tive exchanges at}\\ \text{the indoor surface}\end{array}\right] \quad . \; . \; (1)$$

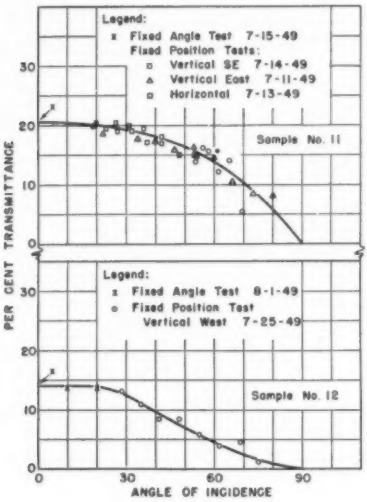

FIG. 2. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY SAMPLES NOS. 11 AND 12

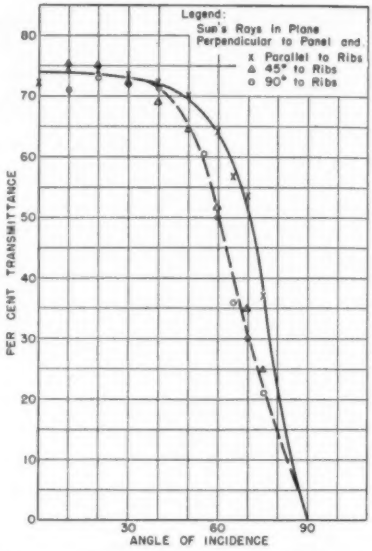

FIG. 3. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY SAMPLE NO. 15

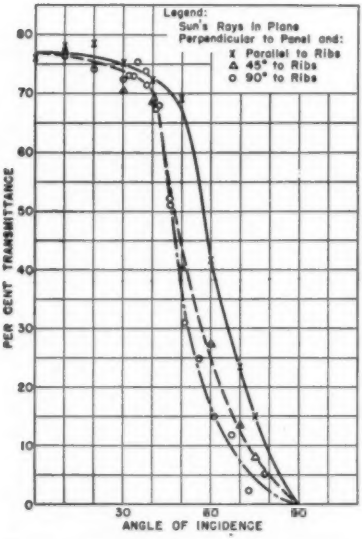

FIG. 4. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY SAMPLE NO. 16

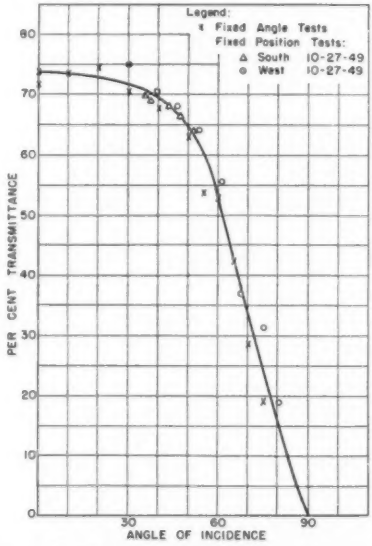

FIG. 5. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY SAMPLE NO. 17

The data on transmittance just presented show that the transmitted solar radiation is dependent upon the glass properties and the nature and angle of the incident solar radiation. On the other hand, the heat flow by convective and radiative exchanges at the indoor surface is dependent upon these same factors

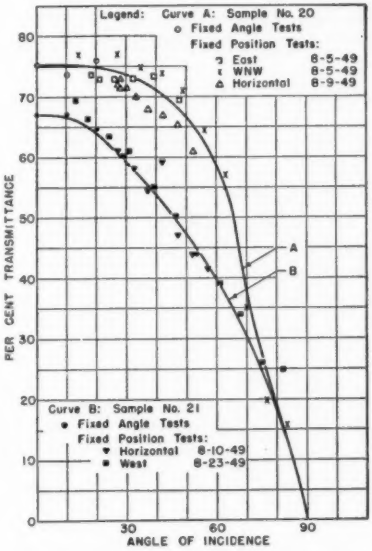

FIG. 6. TRANSMITTANCE OF DIRECT SOLAR RADIATION BY SAMPLES NOS. 20 AND 21

plus the influence of the indoor and outdoor environments. This heat flow can be expressed by a second heat balance equation as follows:

$$\left[\begin{array}{l}\text{Heat flow by convective and radiative exchanges at the indoor surface}\end{array}\right] = \left[\begin{array}{l}\text{Absorbed solar radiation}\end{array}\right] \pm \left[\begin{array}{l}\text{Radiative exchanges between outer surface of glass and outdoor surroundings}\end{array}\right] \pm \left[\begin{array}{l}\text{Convective exchanges between outer surface of glass and outdoor air}\end{array}\right] \pm \left[\begin{array}{l}\text{Heat storage within the glass section}\end{array}\right] \quad \ldots\ldots\ldots \quad (2)$$

Fig. 7 illustrates Equations 1 and 2 by giving instantaneous rates of heat gain for a west-oriented vertical sheet of glass No. 21. Total heat gain and the convection and radiation heat flow from glass to calorimeter as determined by test have been corrected to a zero differential temperature between the outdoor air and the calorimeter surface. The gain due to transmitted solar radiation is the difference between the two heat gain curves.

DISCUSSION OF RESULTS

Solar Energy Transmission: Table 2 shows that the normal incidence transmittance of polished samples is subject to some variation. This is due primarily

to differences in the energy distribution of the incident rays caused by day-to-day and hour-to-hour changes in the thickness and composition of the atmosphere in the sun's path. These variations are somewhat smaller than has been observed in tests on other samples because of the smaller number of tests made.

Fig. 8 is presented in order to show the effect of pattern or surface treatment on transmittance. The figure shows transmittance as a function of incident angle for flat polished sheets of the same overall thickness as the patterned

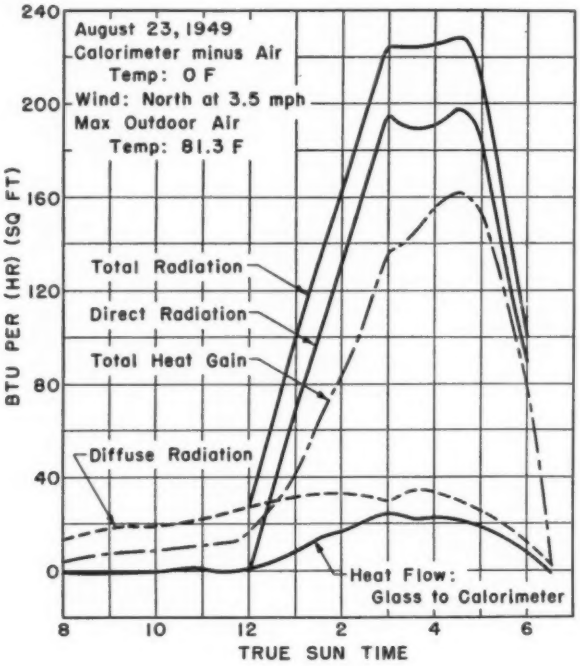

FIG. 7. TEST RESULTS FOR GLASS NO. 21, ORIENTED WEST

sheets. The curves were computed for an energy distribution according to Moon's[5] suggested standard for air mass of two. Comparison of Fig. 8 with Figs. 2 through 6 shows that even the simplest patterns significantly reduce transmittance. The effect of etching to produce glare reducing surfaces is even more marked.

Because the transmittance of the ribbed glasses, Nos. 15 and 16, cannot be given in terms of incident angle only, two sets of iso-transmittance curves are presented in Figs. 9 and 10. These give transmittance as a function of altitude and azimuth for both vertical and horizontal sheets. If the glass is installed with ribs vertical in a vertical wall, the azimuth angle is the angle between the

projection of the sun's rays on a horizontal plane and a line perpendicular to the wall. For horizontal sheets, azimuth is measured with respect to the direction line of the ribs. Solar altitude is, of course, the angle of the sun above the horizon.

Table 3 shows that the transmittance of a glass sheet for diffuse solar radiation may vary widely (the experimental error in the determination of solar gain is of the order of 3 to 5 percent). There is considerable evidence to show

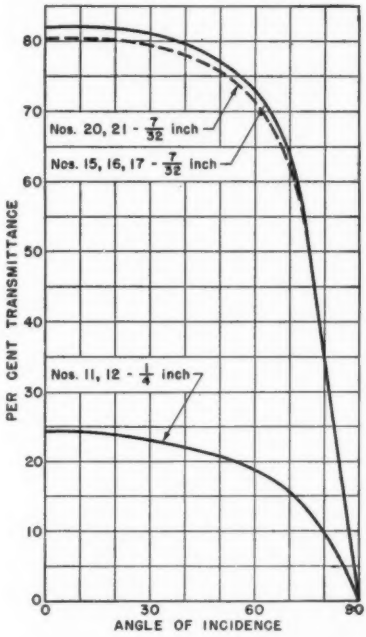

FIG. 8. COMPUTED TRANSMITTANCES OF FLAT POLISHED SAMPLES OF GLASS IDENTICAL IN COMPOSITION WITH FIGURED GLASSES STUDIED

that much of this is probably due to the effects of angular distribution of the incident diffuse radiation.

In order to study the magnitude of this effect, diffuse transmittance tests were made with a 2-ft sq sheet of high transmission glass. Use of this type of glass eliminated the possible influence of energy distribution. The tests were made by placing the glass over the pyrheliometer, observing the emf, and then observing the emf when the glass was removed. The pyrheliometer was shaded from the direct sun at all times by a 3-in. diameter disk about 30 in. away from the instrument.

The investigation showed that the diffuse transmittance varied with the incident angle, being greatest at normal incidence and dropping as the angle was reduced. There was about 8 percent difference between the lowest and highest values. This can be accounted for by the fact that much of the diffuse radiation comes

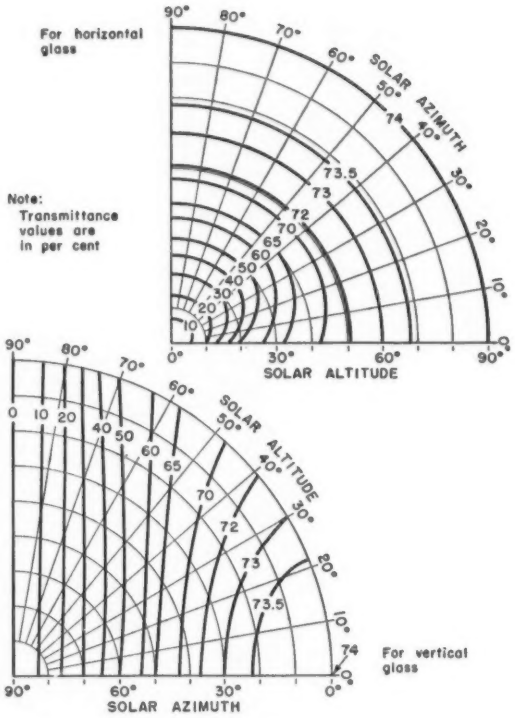

FIG. 9. DIRECT SOLAR RADIATION ISO-TRANSMITTANCE CURVES FOR SAMPLE NO. 15

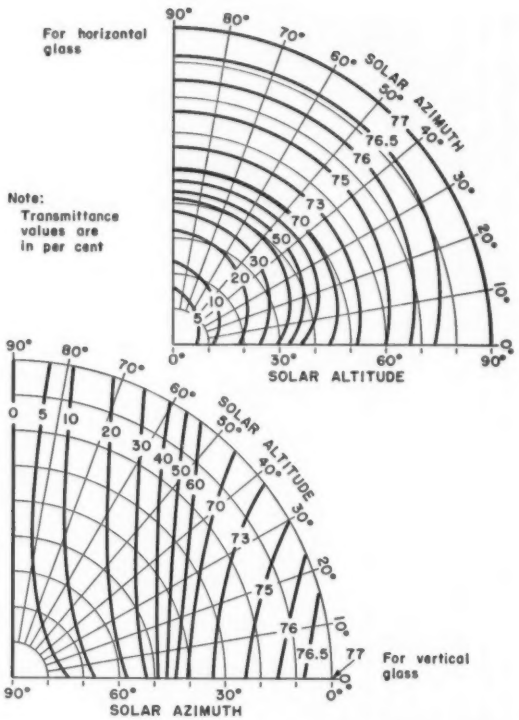

FIG. 10. DIRECT SOLAR RADIATION ISO-TRANSMITTANCE CURVES FOR SAMPLE NO. 16

from the near vicinity of the sun and hence is transmitted in accordance with the incident angle. One would therefore expect the transmittance to vary also with the ratio of direct to diffuse radiation.

The lack of uniformity in the angular distribution of diffuse solar energy is further illustrated by observations of the variation in the diffuse irradiation of vertical and horizontal surfaces with altitude and azimuth. For the same solar altitude it was found that the ratio of diffuse radiation falling on a vertical plane to that on a horizontal varied from about 0.5 with the sun 180 deg away from the wall to as much as 1.4 when the sun's azimuth with respect to the wall was zero. If the diffuse radiation of all points of the sky was of equal intensity, this ratio would be 0.5.

On the average the diffuse transmittances were closely equal to those computed from direct transmittance data and to those found to give the best fit of data. The latter are recommended for design purposes. Two exceptions, the

TABLE 3—PERCENT TRANSMITTANCE OF CONFIGURATED GLASS FOR DIFFUSE SOLAR RADIATION

SAMPLE NO.	BY CALORIMETER TESTS		CALCULATED	DESIGN VALUES
	Range	Av		
11	20.2-25.5	22.4	16.0	20.0
12	12.6-23.6	18.0	8.2	16.0
15	51.2-74.0	60.0	61.6	60.0
16	41.2-58.5	47.9	51.3	48.0
17	49.9-69.8	59.0	59.0	59.0
20	62.6-71.7	67.1	62.9	63.0
21	45.6-59.4	48.3	48.4	48.0

values for heat absorbing glass, were expected. In this case the energy distribution is responsible. Diffuse solar radiation is deficient in infrared energy as compared to that from direct sun; therefore, heat absorbing glass which has a low infrared transmittance will give higher diffuse transmittances than would be predicted from the direct radiation transmittances.

Total Heat Gain: The heat flow data given in Fig. 7 are not to be construed as design data, but simply represent heat flow under the test conditions noted on the figure. The design of the calorimeter is such that instantaneous heat flow rates are instantaneous cooling loads. In most practical cases of air conditioning, much of the transmitted radiation is absorbed by the interior structure and is gradually released to the room air. The resulting apparatus load therefore lags behind periodic instantaneous heat gain and has a reduced amplitude.

The fact that there is considerable radiation and convection gain by the calorimeter from the glass is due to absorption of solar energy. During the morning hours the absorbed solar radiation was very nearly balanced by radiation loss to the outdoor surroundings. This gain was, of course, greater for glasses Nos. 11 and 12 because of their greater absorption.

It is of interest to note that the solar radiation data presented in Fig. 7 very closely approximate the values suggested for a clear atmosphere in a recent research paper[4]. Some departure from the design values occurs in the late

afternoon because sunset on August 23 occurs about ½ hour earlier than on an August 1 design day. Typical of Cleveland, the maximum dry bulb, 81 F, and dew-point 60 F, associated with these solar intensities are much below the usual design values of 95 F and 67 F respectively.

ACKNOWLEDGMENT

The authors acknowledge with thanks the suggestions and assistance given by members of the Technical Advisory Committee on Heat Flow Through Glass and by their colleagues of the A.S.H.V.E. Research Laboratory staff during the course of the work described in this report.

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DISCUSSION

E. B. PENROD*, Lexington, Ky. (WRITTEN): This is an informative paper supplementing the present knowledge of solar energy transmission through glass. It is interesting to note the reduction in direct transmission resulting from surface configuration. However, it also would be of value to know if this reduction occurs as a result of increased absorption or increased reflection. If surface configuration increases the absorption, an appreciable fraction will ultimately be transferred to the inside air by conduction and convection, alleviating the advantage obtained by the reduced transmission.

It is believed that a clarification of the technique used to measure the diffuse component is in order. Since diffuse radiation originates in all directions, how is it possible to establish an angle of incidence?

E. W. CONOVER, Detroit, Mich.: In this paper as in previous ones by the authors, it is stated that glass surface temperatures were measured, but no data are given. These temperatures are significant factors in heat transmission and we believe that the data should be included. Glass temperature enters into the heat loss during daylight hours in winter as well as the cooling load in summer.

The description of figured glass patterns, as given in Table 1, does not seem to be clear enough. Is flint glass the ordinary type of window glass? Are the shallow ribs of No. 15 of the same design as the deep ribs of No. 16, and what are the

* Professor and Head of Department of Mechanical Engineering, University of Kentucky.

spacings? How does No. 17 with small circular indentations differ from so-called hammered glass?

Fig. 1 shows that in the infrared range Samples No. 11 and 12 transmit on the average less than 5 percent. Table 2 for normally incident direct solar radiation shows tests and calculations considerably greater. Is this difference explained by the inclusion in Table 2 of the diffuse solar radiation shown in Table 3? If not, how can Table 2 be reconciled with Fig. 1?

It would appear that, due to the greater surface area of configurated glass, that heat transmission factors for these would be higher than for clear sheet glass. Are these going to be worked out later?

CYRIL TASKER, Cleveland, Ohio: It is perhaps well to put on record a very brief history of this program, since this may be the last paper on this subject that will come from the laboratory.

It is six years since Mr. Parmelee was engaged to survey available information under the auspices of the Technical Advisory Committee on Heat Flow through Glass. From that survey came two programs: one, a relatively short-range program, and another, a long-range program. On the short-range program, dealing with forced convection heat transfer coefficients, work was done, a research bulletin prepared and papers presented before the Society.

The other program dealing with solar radiation transmission, immediately faced us with a great deal of experimental work. Mr. Parmelee and his co-author have presented papers covering the data obtained in the experimental work.

Last summer Mr. Parmelee presented some design heat flow data covering various types of glass. The data here presented will be used as the basis for making up other heat flow design data which we are planning to present.

This year Mr. Parmelee was asked to rewrite the final chapter on cooling load, and incorporate as design data the information developed in these tests. In THE 1951 GUIDE, you will find that the cooling load chapter has in it the sort of information that you can use directly.

Mr. Conover has asked why more data and experimental information are not given? We are hoping to cover that. We realize that it would be quite impossible for Mr. Parmelee to cover in technical papers all the points covered in the course of these studies; we have in preparation another research bulletin to summarize the information collected and give additional information not already presented. This will attempt to tie all this information together and put on permanent record the results of about six years of work.

Mr. Parmelee has paid tribute to the members of the TAC on Heat Flow through Glass. They have been most helpful technically, financially, and in every other way. We wish to say thank you when we come to this rather important stage in the program.

We have a big task ahead of us now in translating all these data on instantaneous heat flow into cooling loads. Two of the committees at this meeting have spent practically the whole day discussing how that might be done. We shall shortly present suggestions for a continuing program for translating instantaneous heat gain into cooling load so that the information will be of maximum use to the air-conditioning industry.

AUTHORS' CLOSURE (G. V. Parmelee): The authors agree with Mr. Conover that glass temperatures are important data that should be given. However, since most of the papers on the glass research which have been presented to date have dealt with solar energy transmittances, we have not stressed glass temperatures. Some information has been given in connection with the results of tests to determine overall coefficients. As Mr. Tasker has indicated we will include considerable information

heretofore unpublished in the summary bulletin now in preparation. This will include glass temperatures.

Flint glass is a term applied to a clear glass, such as ordinary window glass, in contrast to heat absorbing glass, which has a definite tint or color. The shallow ribs of glass No. 15 are spaced 22 to the inch. The included angle of the valley is about 150 deg. Both valley and crest are rounded. Glass No. 16 has ribs spaced 8¼ per inch with an included angle of 110 deg. The pattern of the hammered glass consists of raised approximately circular areas about 0.1 in. in diameter closely spaced in an irregular manner, the result of hammerlike indentations in the rolls. In No. 17 the indentations are circular, 19 to the inch and arranged in straight rows.

The infrared transmittance of samples No. 11 and 12 is of the order of 5 percent but the total solar energy transmittance is 20 to 25 percent, as shown in Table 2. Note that these values are for specimens ground and polished to remove the pattern. This is because only 50-55 percent of the solar energy is in the infrared region, that is, longer than about 0.75 microns. In the visible region, the transmittance is much higher approaching 70 percent at 0.50 microns. When you integrate over the spectrum from 0.3 to about 2.2 microns, you get the results given in Table 2, whether the transmittance is calculated or found by tests. We have pointed out in previous papers that transmittance of the heat absorbing glasses is markedly affected by differences in the energy distribution in the solar spectrum. The transmittances in Table 2 are for normally incident solar radiation. The transmittances of diffusely incident radiation are given in Table 3 for *patterned* glass.

Mr. Conover brought out a good point, namely, the effect of configuration on surface area and heat transmission. This has no bearing on the results presented in this paper. However, in calculating solar absorptions of glass sections we increase the emissivity by a few percent over the value for a smooth sheet to take surface roughness into account. Undoubtedly it has an effect on convection heat transfer, but that is something we cannot determine with our calorimeter apparatus.

Professor Penrod has brought up several interesting points, one of which, absorption, has not been touched upon in any of our papers. By means of heat balances we have worked out absorptions for various glass sections as a function of incident angle. The determination of these values is rather difficult because of the problem of making an accurate heat balance. You need to know, for example, the convection heat transfer coefficient and the long wave length radiation received from the sky. We have gone to some trouble to determine these factors to a satisfactory degree of accuracy in order to determine absorptances.

It is true that if the configuration increases absorption, there will be some heat gain by convection and reradiation from the glass that would otherwise not be present. However, I think in most cases the pattern increases reflectance rather than absorptance.

Professor Penrod asks for clarification of the technique used to measure the diffuse component. I don't know whether he refers to measuring the transmittance of the diffuse solar component or measuring the diffuse component of the solar radiation. We have used two Eppley 180-deg pyrheliometers. Since these have flat receiving elements, they "see" the whole sky. One was exposed to the direct sun and recorded total radiation. The other was shaded from the direct sun by a small disc about 2½ in. in diameter and 30 in. from the pyrheliometer. In this way the pyrheliometer could "see" the sky and all the surroundings except the direct sun. Thus it measured the incident diffuse radiation.

In determining the diffuse transmittance, we simply exposed the glass to diffuse solar radiation only by orienting it away from the sun. This factor was then used to separate the transmitted diffuse radiation from the total transmitted solar radiation, thereby permitting determination of the transmittance of the glass for direct radiation. By using a technique described in the paper we found that incident angle had some effect on diffuse transmittance.

1418

No. 1418

FIELD STUDIES OF HEAT LOSSES FROM CONCRETE FLOOR PANELS

By C. M. HUMPHREYS,* C. V. FRANKS** AND L. F. SCHUTRUM†
CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THE SOCIETY, through the Committee on Research, is engaged in a comprehensive research program on panel heating, the general direction of which is under the Technical Advisory Committee on Panel Heating and Cooling. As a part of this program the Society's Research Laboratory carried on a field study during the 1949-1950 heating season to determine the order of magnitude of heat losses from floor slabs to the earth and to study the effectiveness of insulation in reducing these losses.

Four single story basementless frame houses were used for these tests. The houses are located in Parma Heights, Ohio, and are identical in size and arrangement, except that the floor plans of alternate houses are reversed. An exterior view of the four houses is shown in Fig. 1. All of them were occupied during the period of these studies.

The calculated heat loss for each house, based on inside and outside temperatures of 70 F and 0 F respectively, was 26,500 Btu per hour. In some of the houses, heat losses were reduced during the heating season by the installation of storm windows.

Thermostats in all four houses were set at 75 F, and except in very mild weather, this temperature was maintained within about ± 1 F.

A curve giving degree days throughout the test period is shown in Fig. 6. This curve was plotted from 3-day averages.

PANEL CONSTRUCTION AND INSULATION

The floor panels in all four houses were identical and consisted of 4 in. concrete slabs containing sinuous coils of ½ in. nominal copper tubing spaced 9 in. center

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TABLE 1—SLAB INSULATION

INSULATION INSTALLED BETWEEN	TYPE AND AMOUNT OF INSULATION USED			
	House No. 1	House No. 2	House No. 3	House No. 4
Slab and fill	None	4 in. Hollow Tile	2 in. Cellular Glass Block[a]	2 in. Cellular Glass Block[b]
Slab and top of foundation	None	None	2 in. Cellular Glass Block	2 in. Cellular Glass Block
Slab edge and frame wall	None	None	None	Cellular Glass Block

[a] Under entire house.
[b] 3 ft 6 in. wide border around perimeter of house.

to center. All the slabs were covered with a ⅛ in. thick asphalt tile floor. The slab in house No. 1, which was typical of the standard construction in use at that time, was laid without insulation. In the other three houses insulation was installed between the fill and the slab as indicated in Table 1. A vapor barrier of 30 lb saturated slaters felt was provided between the slab and the gravel or the insulation.

Details of insulation installation are given in Fig. 2. As shown in the drawings, the gravel fill under the houses was unusually deep.

INSTRUMENTATION

As soon as the concrete foundations were poured to rough grade, work was started on the installation of thermocouples and heat flow meters. Five groups of thermocouples, designated by letters A to E inclusive, were installed in and under the floor slab of each house. Their approximate locations are shown in plan view in Fig. 3. The vertical distribution of the thermocouples in each group is shown in Fig. 4.

Group A includes thermocouples 1 to 11 inclusive, which were located along the inside and outside of the foundation wall. Couples 1, 2, 5 and 6 were imbedded in the mortar joints as the brick portion of the foundation was laid.

FIG. 1. VIEW OF TEST HOUSES

Other couples in this group were installed in holes along the concrete portion of the foundation.

Thermocouple groups B and D extend to depths of approximately 8 and 10 ft, respectively, below the top of the floor slab. These groups of couples differed slightly in the four houses as required by the different slab insulations. The couples below rough grade were installed by attaching them to a wood pole ½ in. square and placing the pole in a hole bored into the earth with a 2 in. auger. The hole was then filled with dry sand.

Groups C and E each consist of a heat flow meter and four thermocouples, one of which is in contact with the meter. These were installed to obtain a direct method of measuring heat flow and to provide a means of determining

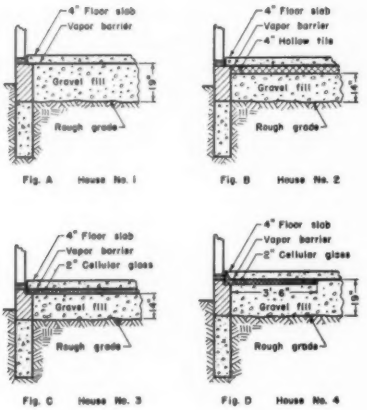

FIG. 2. DETAILS OF FLOOR INSULATION

the thermal conductivity of the earth. The thermocouples under the meters were installed without disturbing the earth.

Leads from the thermocouples and heat flow meters in each house were run under the gravel fill to the garage, where a suitable switching panel was installed.

Special care was taken to protect underground and concrete-imbedded wiring from moisture damage. All thermocouple wires and heat flow meter leads were given two coats of a waterproof lacquer. Each thermocouple and its leads were then encased in a small plastic tube, the bottom of which was heat-sealed. The heat flow meters were given several dip coats of a heavy asphalt paint. These precautions were warranted for there was no evidence during the tests of any failures due to moisture. It was found when the thermocouples were first connected that both heat flow meters and one thermocouple on house No. 1 were dead, but this was undoubtedly the result of mechanical damage.

All thermocouples were made of No. 26 copper and constantan wire. No. 26 gage copper wire was also used for the heat flow meter leads. A portable

precision potentiometer was used to take thermocouple and heat flow meter readings.

COLLECTION OF DATA

Readings of all thermocouples and heat flow meters in each home were taken at approximately two-week intervals from October 5 to May 31. On each visit the air temperature was also taken in the living room near the thermostat located at the 60 in. level.

Since temperatures and heat flow rates were found to be subject to constant variation, a 16-point electronic recording potentiometer was installed in each

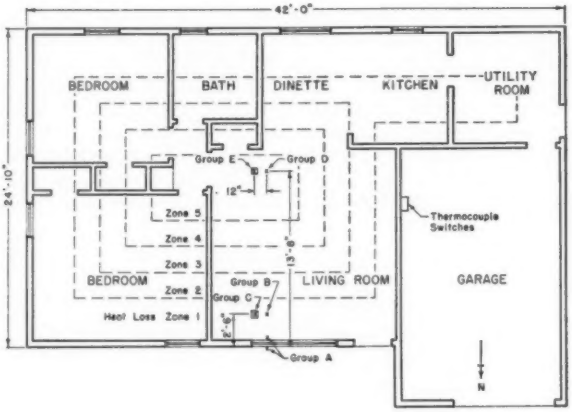

FIG. 3. TYPICAL FLOOR PLAN SHOWING APPROXIMATE LOCATION OF METERING EQUIPMENT

of the houses for approximately two weeks to obtain a complete record of these cyclic effects. Continuous records of the two heat flow meters and 14 selected thermocouples were thus obtained.

Additional data on edge losses were obtained for each house by temporarily attaching a heat flow meter to the exterior of the foundation wall above grade and connecting it to the recorder for a few days in place of one of the thermocouples.

SOIL ANALYSIS

A test boring was made between houses Nos. 3 and 4, and mechanical analyses were made of the soil samples removed from each foot of depth. The results of these analyses are shown in Table 2.

RESULTS

The downward heat flow at any point under a floor panel is dependent upon a complex temperature pattern. This pattern varies with the continually chang-

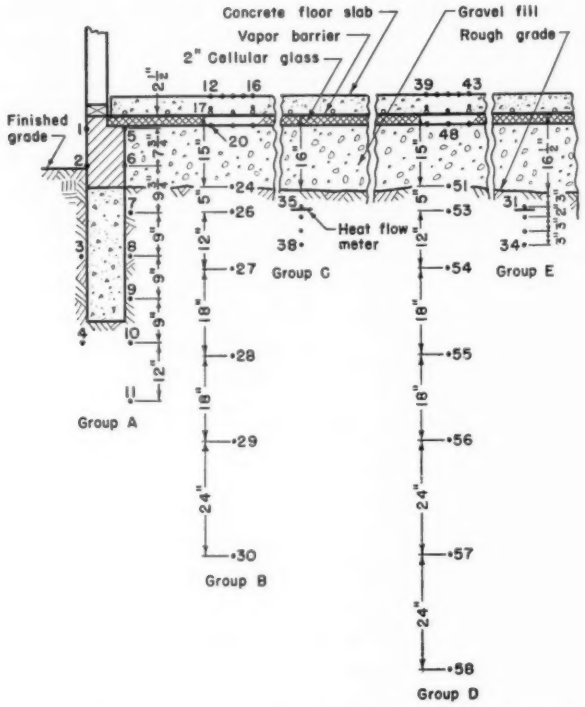

FIG. 4. SECTION SHOWING LOCATION OF THERMOCOUPLES IN HOUSE NO. 3

ing outdoor temperature, the cyclic operation of the heating system, and the thermal conductivity of the soil. Even the soil conductivity may vary from week to week with changes in moisture content.

The cyclic effect of the intermittent operation of the heating system is illustrated in Fig. 5. Although the amplitude of the variation at the tube is relatively

TABLE 2—SOIL PROPERTIES

DEPTH BELOW ROUGH GRADE FT	PERCENTAGE OF DRY WEIGHT		MOISTURE CONTENT PERCENTAGE OF DRY WEIGHT
	Gravel and Sand	Silt and Clay	
1	22.4	77.6	13
2	23.6	76.4	14
3	27.8	72.2	15
4	36.3	63.7	14
5	39.0	61.0	not determined

large, its period is of such short duration that the effect is quite shallow. In contrast to this, a change in weather from mild to cold or vice versa will produce a cycle of relatively low amplitude, but because of its much greater period, its effect may extend to a considerable depth below the slab.

Fig. 6 shows the variation in earth temperatures under house No. 2 throughout the heating season. These temperatures were taken by a group of thermocouples (Group D, Fig. 3) located near the center of the house. It will be noted that the temperature at a depth of 4 ft 6 in. remained approximately constant throughout the year. Above this level the temperatures show a gradual rise until about March 1. Temperatures below the 4 ft 6 in. level reached a peak about December 1 and showed a steady decrease from that date to the end of the heating season.

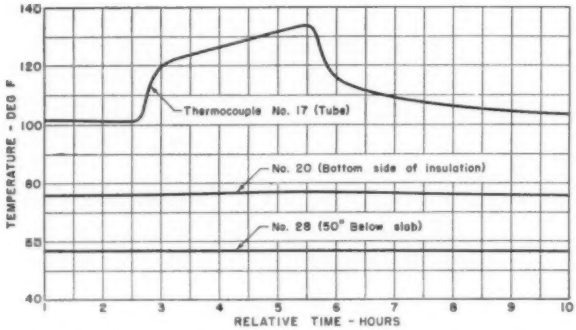

FIG. 5. TYPICAL TEMPERATURE CYCLE FROM HOUSE NO. 3

The same decrease in deep soil temperatures is shown in Fig. 7 by the progressive rise of the 65 F isotherm under house No. 2. The effect of outside air and ground temperatures on ground temperatures just inside the foundation is also indicated by the shape of the isotherms near the wall.

In a similar study[1] by A. B. Algren, ground temperatures to a depth of 6 ft below the slab increased steadily during the heating season.

In keeping with the temperature variations discussed previously, heat flow meter readings indicated variations in heat flow rates from week to week. Fig. 8 shows results for the season obtained from heat flow meters in house No. 2. It will be noted that there was a general increase in heat flow until approximately March 1. This is consistent with the data presented in Fig. 6 which indicate an increasing difference between the shallow and deep ground temperatures until approximately the same date.

Because of the many uncontrollable variables involved, downward heat losses from the panel were computed from average seasonal values. Figs. 9 and 10 show the average seasonal isotherms under houses Nos. 2 and 3, respectively.

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[1] Ground Temperature Distribution with a Floor Panel Heating System, by A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 321).

It will be noted that ground temperatures under the insulated slab in house No. 3 are lower, and that the isotherms are farther apart, than in house No. 2.

On Figs. 9 and 10, the direction of heat flow at two locations under the slab is shown by dotted lines drawn normal to each of the isotherms crossed. The heat flow line noted D at the left of each figure passes through the bottom of the foundation. It is evident from the figures that all of the downward heat flow from that part of the slab between the foundation and the intersection with

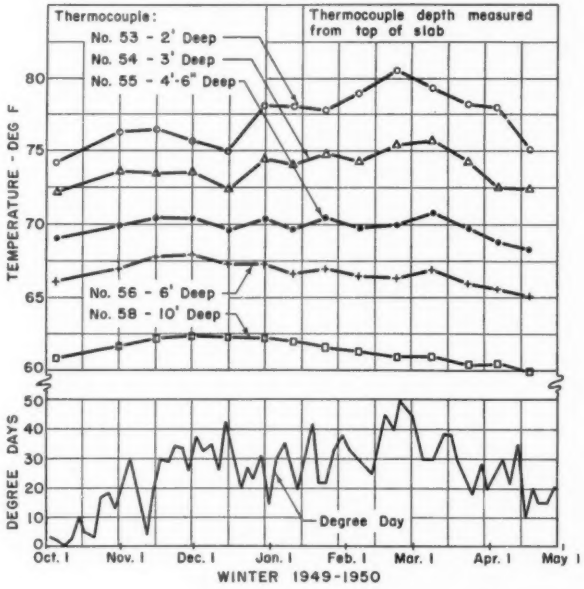

FIG. 6. SUB-SLAB TEMPERATURE VARIATIONS AT CENTER OF HOUSE NO. 2

line D must pass through the foundation. The loss for this part of the slab (heat loss zone No. 1 shown by dotted lines on Fig. 3) was calculated from the thermal conductivity of the foundation and the temperature drop across it.

Fig. 11 shows the average seasonal temperature drop across the foundation walls for all four houses. The effect of the edge insulation in houses Nos. 3 and 4 is also evident.

The center of the floor slab was arbitrarily divided into several zones of approximately uniform heat loss as shown in Fig. 3. The heat loss from each of these zones was calculated from the soil thermal conductivity and the spacing of the isotherms.

The average thermal conductivity of the soil under houses Nos. 2, 3 and 4 was calculated from the heat flow meter readings and the soil temperatures at

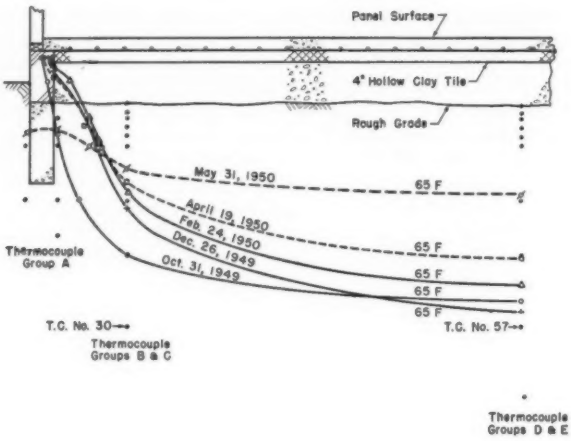

FIG. 7. VARIATION IN LOCATION OF THE 65 F ISOTHERM UNDER HOUSE NO. 2

known distances from the meters. Since the flow meters in house No. 1 were inoperative, the conductivity of the soil under it was assumed as the average of those determined for the other three houses. The values so determined agreed well with the thermal conductivity of (6.6 ± 25 percent) Btu per (hr) (sq ft) (deg F per inch) obtained on the basis of mechanical soil analysis from a recent bulletin[2] from the University of Minnesota.

A summary of the average ground heat losses for the four houses for the period from Oct. 5 to Apr. 19 is given in Table 3. A comparison of the losses from the several zones indicates that the major part of the loss from a floor

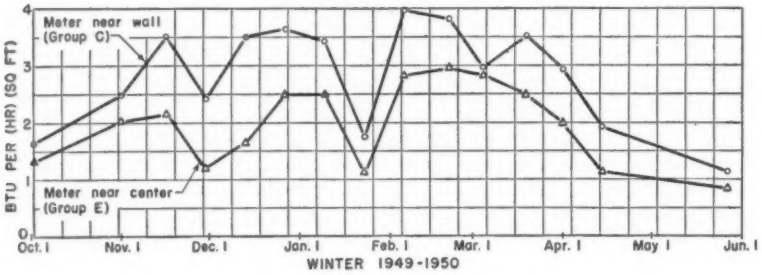

FIG. 8. HEAT FLOW RATES FROM HEAT FLOW METERS IN HOUSE NO. 2

[2] Thermal Properties of Soils by Niles S. Kersten, (*Bulletin No. 28*, Engineering Experiment Station, University of Minnesota).

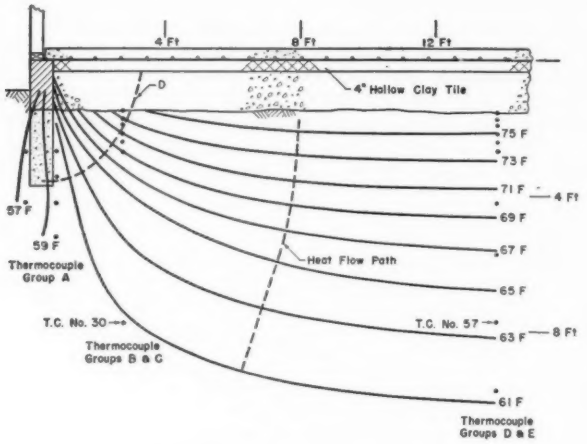

FIG. 9. AVERAGE SEASONAL ISOTHERMS UNDER HOUSE NO. 2

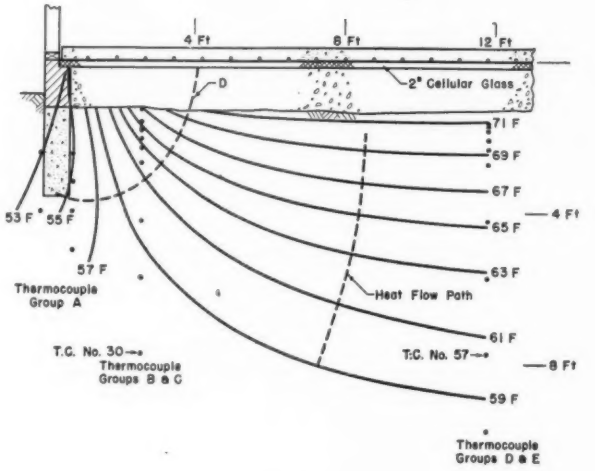

FIG. 10. AVERAGE SEASONAL ISOTHERMS UNDER HOUSE NO. 3

panel takes place from a comparatively narrow band around the edge. It is in this area that insulation will pay its greatest dividends.

Total gas consumption for each house was obtained from gas company records. These total consumption values were corrected on the basis of the average summer consumption for each house, to obtain the net Btu input to each of the heating systems. The Btu output of each boiler from October 5, 1949 to April 19, 1950 was calculated on the assumption of 75 percent efficiency and compared to slab losses for the same period. The ground heat losses, expressed as a per-

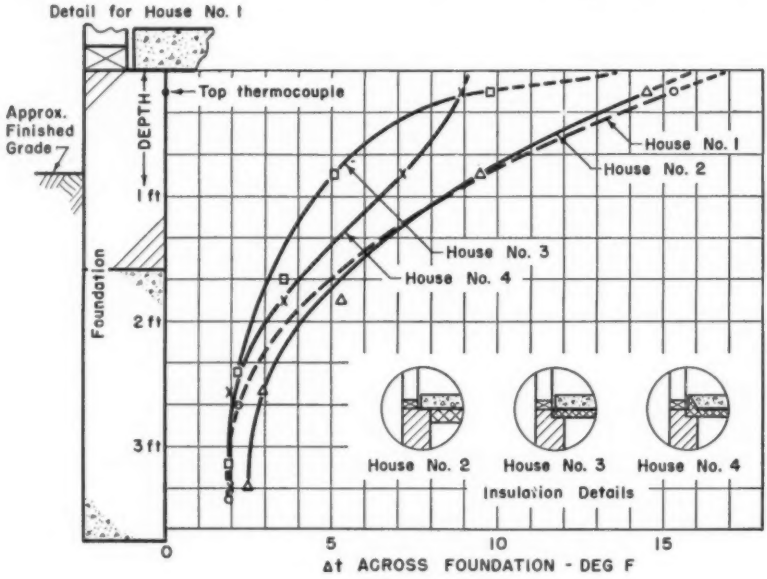

FIG. 11. AVERAGE SEASONAL TEMPERATURE DROP ACROSS FOUNDATION

centage of the boiler output, averaged approximately 33 percent for houses Nos. 1 and 2, and approximately 25 percent for houses Nos. 3 and 4.

DISCUSSION OF RESULTS

Data have been presented on the heat losses from floor panels in four houses. When examining these data it is well to consider some modifying factors which, although they cannot be evaluated, do have a material effect on the heat losses.

It has long been recognized that the type of occupancy can materially affect fuel consumption and heating system performance. Although the occupancy was a factor over which the Laboratory had no control, it was hoped that it would be similar in all houses. Actually it varied so widely that it is considered worthy of note.

House No. 1 was occupied by a couple, both of whom worked. They were home only in the evenings. In house No. 2 were a man and his wife and two

small children. An unmarried nurse purchased house No. 3. The house was sometimes unoccupied for a week or two at a time while she was on a case. House No. 4 was occupied by a man and his wife; the wife was usually at home.

Floor coverings can have an appreciable effect on the downward heat loss from a heated floor slab. Carpets or rugs act as an insulation on the top of the panel and cause a greater heat flow downward. During the period in which

TABLE 3—HEAT LOSSES FROM FLOOR SLABS

ZONE NOS.	HOUSE NO. 1	HOUSE NO. 2	HOUSE NO. 3	HOUSE NO. 4
	Zone Areas, sq ft			
1	362	356	422	463
2	199	199	179	163
3	120	121	110	104
4	88	89	78	72
5	71	75	51	38
	Average Heat Losses, Btu/(sq ft) (hr)			
1	8.53	8.88	4.71	4.44
2	3.05	3.10	1.50	3.35
3	1.95	2.10	1.10	2.90
4	1.62	1.80	1.00	2.60
5	1.50	1.70	1.00	2.20
	Total Average Zone Losses, Btu/hr			
1	3090	3160	1990	2057
2	609	617	268	547
3	234	255	121	302
4	142	161	79	187
5	107	128	51	84
	Total Average Losses from Panels, Btu/hr			
	4182	4321	2509	3177
Ratio[a]	0.43	0.425	0.50	0.55
Ratio[b]	0.74	0.73	0.795	0.65

[a] Area of zone No. 1/total floor area.
[b] Heat loss from zone No. 1/total heat loss from panel.

these tests were being made, floor coverings varied, not only from house to house, but from time to time in the same house.

There were also at least two unusual factors which tended to reduce the heat losses from all houses, and to minimize the differences between the insulated and uninsulated installations. One of these was the abnormally deep (14 to 19 in.) gravel fill between the rough grade and the slab. The other was the unusually mild weather experienced last year in the Cleveland district.

ACKNOWLEDGMENTS

The Committee on Research and the authors wish to express their appreciation to the Precision Housing Corp. for making the four houses available for

test and installing the insulation and tile under the slabs, to the Pittsburgh Corning Corp. for the insulation used in houses 3 and 4, to the *Structural Clay Tile Products Association* for the tile used under the slab in house 2, and to the occupants of the four houses for their ready cooperation in making the homes accessible to members of the Laboratory staff for taking data.

CONCLUSIONS

1. The greatest part of the heat loss from a floor panel occurs around the perimeter of the panel. It is in this area that insulation will prove most effective.
2. It is particularly important that the panel be separated from the foundation by suitable insulation.
3. Four inch hollow clay tile does not appear to be any more effective as an insulation under a floor panel than an equivalent thickness of gravel. Its use as an edge insulation was not studied.

DISCUSSION

M. J. GIBBONS, Dayton, Ohio: What is the most satisfactory type of insulation for insulating the edge?

AUTHORS' CLOSURE (L. F. Schutrum): Since only one type of edge insulation was used in the tests, we are unable to answer Mr. Gibbons' question. However, it should be emphasized that our purpose in these tests was to determine the effect of insulation in general, not to compare specific insulating materials.

1419

No. 1419

PRINCIPLES OF EFFECTIVE HEAT FOR STEAM-HEATED STRUCTURES

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EFFECTIVE heat is a term which has been used in the literature of heating, as the references indicate, for at least 25 years. The accepted meaning of this term will be explained before a precise definition is offered. Assume that it is desired to maintain a certain air temperature, 68 F, in the center of a room at a level of 24 in. above the floor, when the room is exposed to fixed conditions outside all surfaces. Each of several heating devices, whether radiator, convector, or baseboard, may maintain this air temperature *at the fixed point* under equilibrium conditions. In each case, however, it is possible that the rate of heat emission of the heating device may be different. With equilibrium conditions and no storage of energy, the rate of heat loss from the room must always be the same as the rate of heat emission within the room. There are at least two influences which may cause the rate of heat loss from the same room to be different even though the air temperature at a given point in the room is the same and the conditions affecting heat transfer on the outside of the room are not changed.

One of these influences arises from the fact that the air temperature within a room is not the same at every point. It is well known that the air temperature near the ceiling of a heated room is greater than the air temperature near the floor; the magnitude of the temperature gradient being dependent, in part, upon the type of heating device used. A device that produces a large ceiling to floor

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air temperature gradient may cause a larger heat loss from the room than a device producing a small gradient, even though each unit maintains the same air temperature at some *low* level in the room. This effect may be found even in a room with a perfectly insulated ceiling and floor, because the average air temperature causing heat transfer through exposed vertical surfaces may be greater. Of course, if the object of heating were to maintain some arbitrary, fixed air temperature at a *high* point in the room, the device creating the greater ceiling to floor gradient might operate with smaller heat loss.

The second influence is the local and direct effect that a heating device may have upon heat loss through the structure. It is entirely possible to test two heating devices in the same room and discover that the rates of heat emission

TABLE 1—MAXIMUM ADDITIONS TO CAPACITY FOR CONVECTORS WITH FRONT OUTLETS

OUTLET HEIGHT, MEASURED FROM FLOOR TO TOP OF OUTLET, IN.	ADD TO CAPACITY PERCENTAGES NOT EXCEEDING THOSE LISTED BELOW TO ARRIVE AT RATING, PERCENT
36 or more	0
34	1
32	2
30	3
29	4
28	5
27	6
26	7
25	8
24	9
23	10
22	11
21	12
20	13
19	14
18 or less	15

are different even though the *average* temperature of the air *at all levels in the center of the room* is the same. It is also possible to obtain the more puzzling combination of lower rate of heat loss from the room and greater average temperature of the air at all levels in the center of the room. These results are explained on the basis of a different local effect of the heating unit on the rate of conduction of heat through the wall near the unit.

An experimental method of measuring effective heat must be based upon the following procedure:

1. Select some heating device as the standard with which other heating devices are to be compared; for example, the standard device might be cast-iron radiators of the same design and height with different numbers of sections.

2. Choose the location of the index point and the property that is to be measured at this point as an indication of comfort; for example, this index point might be arbitrarily taken at the 24-in. level in the center of the room, and the air temperature might be selected as the property to be measured.

3. With a fixed temperature in the cold space, install the standard heating units and measure rates of heat emission and corresponding air temperatures at the index

point; from these results, a graph may be plotted with temperature difference (air temperature at index point minus cold space temperature) *vs.* rate of heat emission; this procedure is better than trying to hold the air temperature at the index point constant in all tests by wrapping or blocking portions of the heating unit, as the latter alters the heating unit.

4. Install the non-standard heating units in the same test room and, for the same cold space temperature used in the standard unit tests, measure the rates of heat emission and the corresponding air temperatures at the index point.

Heating effect may now be calculated in the following way: assume that the non-standard unit shows a rate of heat emission of q Btu per hour with a tem-

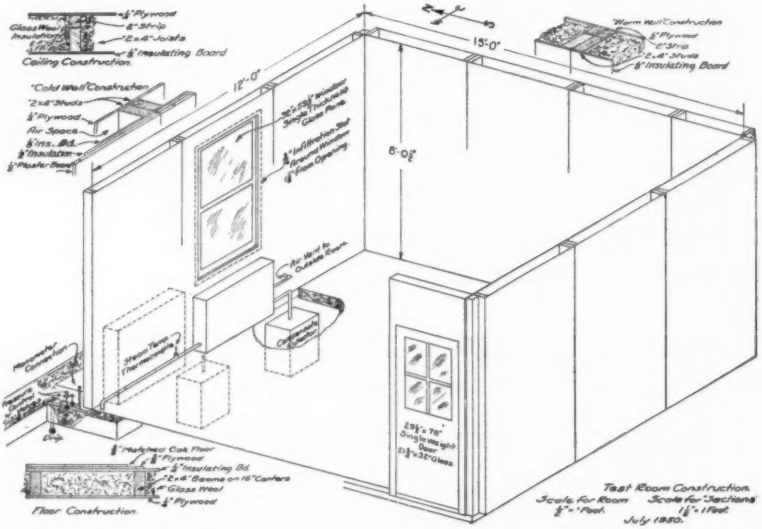

FIG. 1. TEST ROOM CONSTRUCTION

perature difference of Δt between air at the index point and cold space air. From the graph, plotted as a result of the tests of the standard units, read the rate of heat emission, q_s, of the standard unit required to maintain the same temperature differential. Then, percent effective heat is

$$EH = 100\ (q_s/q - 1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In short, effective heat shows how much greater the rate of heat emission of the standard heating unit must be than that of the non-standard unit in order to maintain the same air temperature at the index point in the test room with the same external conditions.

Such a definition gives effective heat as precise a meaning as may be assigned to many such engineering factors determined by tests. It must be stated, at once,

that the same non-standard heating unit, when located in different positions in the same test room, may give different values for effective heat. It must also be clear that standard and non-standard heating units, when tested in test room A, may give different values of effective heat than when both are tested in test room B. This is no new problem to be viewed with great alarm, however, because similar effects may be observed in capacity tests.

HISTORICAL

In 1925, R. V. Frost[1] reported tests on bare and enclosed radiators showing that the enclosed radiators warmed up the air in the test room faster and with a smaller total condensation than did the bare radiators.

In 1926, Dr. C. W. Brabbée[2] commented that the determination of the rate of condensation of steam in a heating unit is not a complete indication of the

TABLE 2—CALCULATED VALUES OF THE OVERALL COEFFICIENTS OF HEAT TRANSFER OF THE CORNELL COLD ROOM

	AREA OF ROOM-SIDE SURFACE, SQ FT A	OVERALL COEFFICIENT OF HEAT TRANSFER, BTU/(HR) (SQ FT) (F DEG) U	RATE OF HEAT TRANSFER BTU/(HR) (F DEG) UA
Net north wall	83.3	0.2	16.7
North window	13.2	1.1	14.5
Net west wall	104.6	0.2	20.9
West door	16.0	1.1	17.6
South wall	96.5	0.045	4.3
East wall	120.6	0.045	5.4
Floor	180	0.067	12.0
Ceiling	180	0.045	8.1
Totals	794.2	....	99.5

effectiveness of that unit in creating a comfortable physical environment, although it is a sufficient indication of the performance of the unit as a steam condenser. He drew a parallel between power input to a lighting fixture as measured by watts *vs.* illumination as measured by foot-candles and rate of condensation of steam *vs.* air temperature at knee height.

In 1927, Dr. Brabbée[3] reported tests on radiators located in two similar rooms showing that one radiator might condense less steam per hour while maintaining a higher air temperature at knee height than another radiator located in a similar room.

In 1928, Dr. Brabbée[4] proposed that radiators be tested in rooms of identical construction exposed to a cold space maintained at constant temperature; one radiator, the standard, was to be kept unchanged in all tests; the number of sections in the other radiator to be tested was selected so that the air temperature near the center of the room and 18 in. above the floor was the same in each room; both radiators were to be supplied with steam in the same state, and

[1] Exponent numerals refer to references.

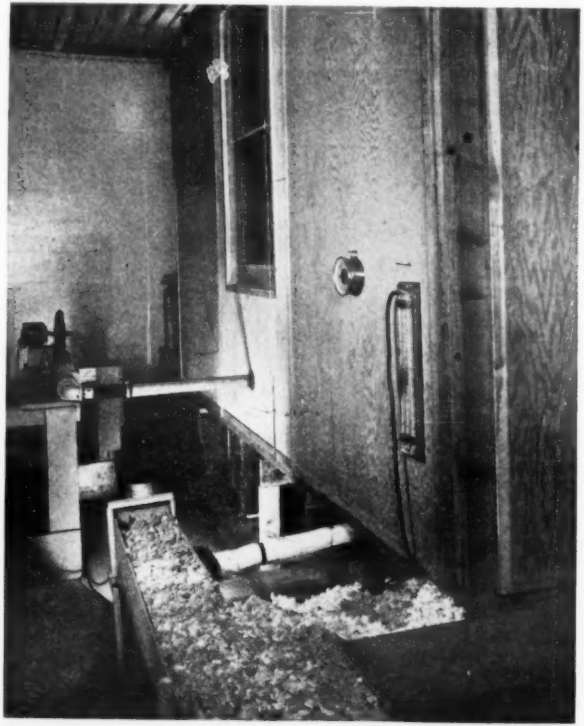

FIG. 2. SHELL SPACE AND NORTH WALL OF COLD ROOM

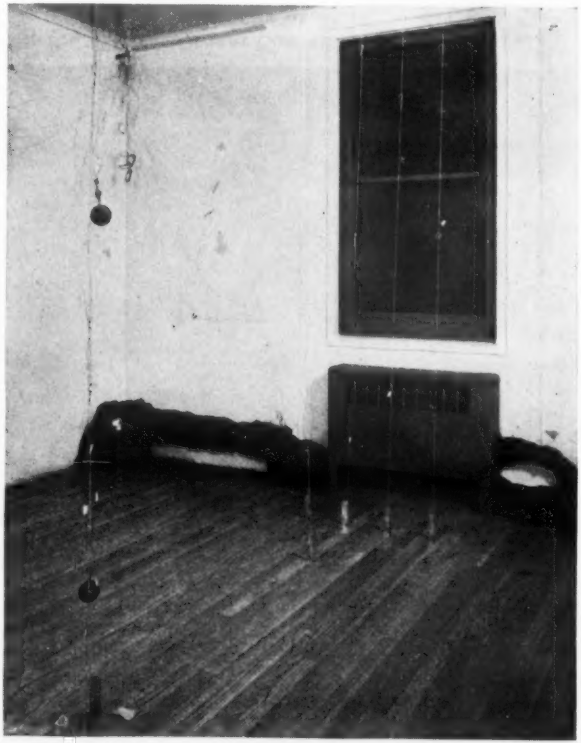

FIG. 3. FRONT OUTLET CONVECTOR WITH ENCLOSURE HEIGHT OF 24 IN.

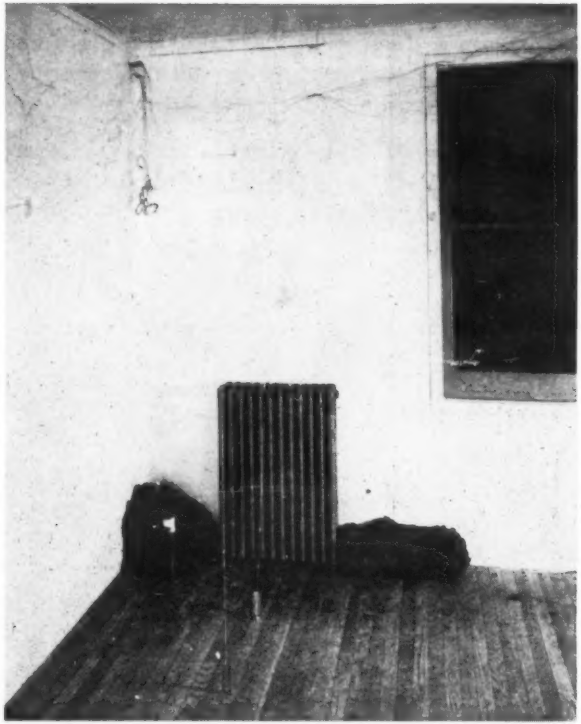

FIG. 4. STANDARD CAST IRON RADIATOR ON NORTH WALL

FIG. 5. TOP OUTLET CONVECTOR WITH ENCLOSURE HEIGHT OF 32 IN.

the rates of condensation were to be compared as a measure of the effectiveness of heating. In the discussion of this paper, Mr. Houghten, Dr. Carrier, and others pointed out that the effectiveness of heating depended not only on the design of the heating unit but also upon the location of that unit and the room in which it might be tested.

In 1929, Willard, Kratz, Fahnestock, and Konzo[5] reported tests on radiators with and without enclosures in 9 ft by 11 ft test rooms with a ceiling height

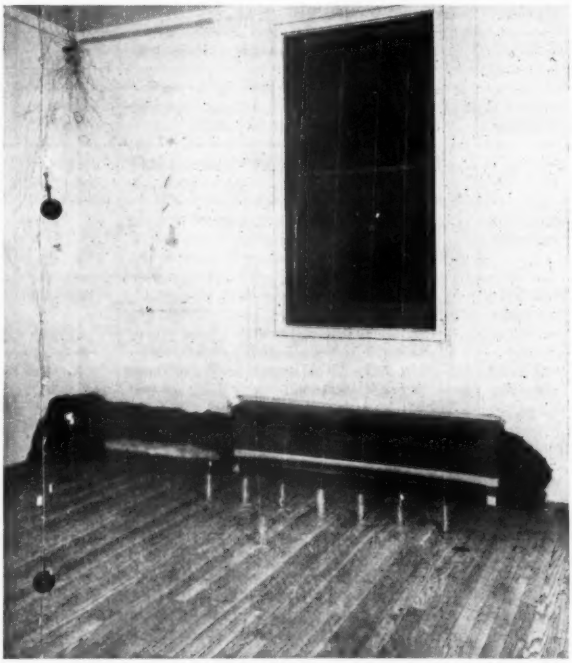

FIG. 6. FRONT OUTLET CONVECTOR; HEIGHT OF TOP OF ENCLOSURE OF 13½ IN.

of 9 ft. The walls of the test room were of frame construction, and cold space temperatures of the order of —1.5 F were used. In discussing the test results, R. N. Trane concluded that: (1) one radiator may keep a room equally comfortable in the living zone while condensing less steam than another; (2) all shields or covers used in the tests lowered the ceiling temperature below that observed with the bare radiator and delivered more effective heat; by calculating the rate of steam condensation per degree of temperature difference between room air at various levels and cold space air, Mr. Trane also showed that one of the enclosures used 13 percent less steam per degree of temperature differ-

TABLE 3—EXPLANATION OF TESTS

TEST NUMBER	UNIT TESTED	DESCRIPTION AND LOCATION OF UNIT	HEIGHT OF TOP OF DISCHARGE GRILLE, IN.	INFILTRATION AROUND WINDOW, CFM
1	7-section C. I. Radiator	32 in. high; between window and west wall	—	12
2	8-section C. I. Radiator	32 in. high; between window and west wall	—	12
3	9-section C. I. Radiator	32 in. high; between window and west wall	—	12
4	10-section C. I. Radiator	32 in. high; between window and west wall	—	12
5	12-section C. I. Radiator	32 in. high; between window and west wall	—	12
11	Front outlet convector	32 in. high; 30 in. long, 6 in. deep; between window and west wall; element 6 in. x 29⅝ in. and 5½ in. above floor at low end	30½	12
12	Front outlet convector	24 in. high; 44 in. long, 4 in. deep; under window; element 4 in. x 43⅝ in. and 5½ in. above floor at low end	22½	12
13	Front outlet convector	24 in. high, 32 in. long, 6 in. deep; under window; element 6 in. x 31⅝ in. and 5½ in. above floor at low end	22½	12
14	Front outlet convector	24 in. high, 26 in. long, 8 in. deep; under window; element 8 in. x 31⅝ in. and 5½ in. above floor at low end	22½	12
15	Front outlet convector	20 in. high, 36 in. long, 6 in. deep; under window; element 6 in. x 35⅝ in. and 5½ in. above floor at low end	18½	12
16	Front outlet convector	12 in. (nominal) high, 54 in. long, 6 in. deep; under window; actual height of 13½ in., entire unit raised so that low end of element is 5½ in. above floor; inlet taped to give same air opening as in test 17; element 6 in. x 53⅝ in.	13	12
17	Front outlet convector	12 in. high; 54 in. long, 6 in. deep; under window; element 6 in. x 53⅝ in. and 4 in. above floor at low end	11½	12
18	Type RC baseboard	10 in. high and 9 ft long; under window on north wall	9½	12
19	Finned tube baseboard with enclosure	8½ in. high and 10 ft long; under window on north wall	8	12

TABLE 3—EXPLANATION OF TESTS *(Concluded)*

TEST NUMBER	UNIT TESTED	DESCRIPTION AND LOCATION OF UNIT	HEIGHT OF TOP OF DISCHARGE GRILLE, IN.	INFILTRATION AROUND WINDOW, CFM
21	30° slant top convector	32 in. high, 27 in. long, 6 in. deep; between window and west wall; element 6 in. x 26⅝ in. and 5½ in. above floor at low end	31½	12
22	30° slant top convector	24 in. high, 32 in. long, 6 in. deep; under window; element 6 in. x 31⅝ in. and 5½ in. above floor at low end	23½	12
23	30° slant top convector	20 in. high, 32 in. long, 6 in. deep; under window; element 6 in. x 31⅝ in. and 5½ in. above floor at low end	19½	12
24	30° slant top convector	12 in. (nominal) high, 54 in. long, 6 in. deep; under window; element 6 in. x 53⅝ in.; actual height of 13½ in. with same air inlet opening as in test 16; low end of element 5½ in. above floor	13	12
31	Top outlet convector	32 in. high, 27 in. long, 6 in. deep; between window and west wall; element 6 in. x 26⅝ in. and 5½ in. above floor at low end	32	12
32	Top outlet convector	24 in. high, 32 in. long, 6 in. deep; under window; element 6 in. x 31⅝ in. and 5½ in. above floor at low end	24	12
33	Top outlet convector	20 in. high, 30 in. long, 6 in. deep; under window; element 6 in. x 29⅝ in. and 5½ in. above floor at low end	20	12
34	Top outlet convector	Same unit as in test 33	20	0
35	Top outlet convector	12 in. (nominal) high, 54 in. long, 6 in. deep; under window; element 6 in. x 53⅝ in.; actual height of 13½ in. with same air inlet opening as in test 16; low end of element 5½ in. above floor	13½	12
36	Top outlet convector	Same unit as in test 35	13½	0
41	Non-Standard C. I. Radiator	19 in. high; between window and west wall	—	12
42	Non-Standard C. I. Radiator	19 in. high; under window; same unit as in test 41	—	12

TABLE 4—SUMMARY OF RESULTS OF TESTS OF CAST IRON RADIATORS USED AS STANDARDS

	TEST NUMBER				
	1	2	3	4	5
Air Temperatures, F:					
Center of room:					
3 in. above floor	59.8	64.2	67.1	70.0	75.3
6 in. above floor	60.3	64.8	68.0	70.7	76.2
12 in. above floor	61.6	66.1	69.4	72.4	78.0
18 in. above floor	61.7	66.2	69.6	72.7	78.3
24 in. above floor	62.4	67.3	71.0	74.3	79.9
30 in. above floor	63.6	68.6	72.5	75.7	81.9
36 in. above floor	64.9	70.3	74.0	77.8	84.3
48 in. above floor	68.7	74.6	78.3	82.5	89.4
60 in. above floor	73.2	78.9	82.4	87.0	94.2
72 in. above floor	75.4	81.5	85.4	89.6	96.1
93½ in. above floor	79.9	86.7	91.1	95.5	102.4
12 in. from west wall:					
3 in. above floor	58.9	63.7	67.0	70.2	74.9
6 in. above floor	59.5	64.1	67.3	70.8	75.6
12 in. above floor	59.8	64.7	67.8	71.3	76.4
18 in. above floor	60.5	65.5	68.9	72.5	78.0
24 in. above floor	62.0	67.1	70.3	74.6	79.8
30 in. above floor	63.4	68.4	71.8	75.7	82.1
36 in. above floor	65.4	70.5	74.2	77.5	84.5
48 in. above floor	69.7	75.2	78.3	82.9	89.8
60 in. above floor	72.7	78.4	81.5	86.3	93.3
72 in. above floor	76.3	82.3	85.5	90.0	96.4
93½ in. above floor	82.1	88.8	93.0	97.0	104.3
2½ in. above floor:					
18 in. in front of radiator	61.1	65.9	68.6	72.6	77.3
36 in. in front of radiator	59.6	64.3	68.3	70.8	75.9
Cold space near cold walls	14	14	14	14	14
Globe thermometer (center of room):					
12 in. above floor	61.7	66.3	69.6	73.1	78.5
48 in. above floor	67.0	72.7	76.3	80.2	86.6
Total rate of condensation, lb/hr	6.34	7.00	7.50	8.02	8.96
No load condensation, lb/hr	0.11	0.11	0.11	0.11	0.11
Net rate of condensation, lb/hr	6.23	6.89	7.39	7.91	8.85
Net rate of heat emission, Btu/hr	6033	6672	7156	7660	8570

ence at the 5 ft level and 15.1 percent less steam per degree of temperature difference in the *living zone;* furthermore, these results were obtained with a greater air temperature gradient from floor to ceiling (23 F) with the enclosure than with the bare radiator (21 F).

In 1931, Mr. Frost[6] reported that enclosed radiators gave higher air temperatures and lower condensation rates with substantially the same average air temperature for the full height of the test room than did bare radiators when both were tested in rooms with water-cooled steel walls.

In 1931, Kratz and Fahnestock[7] concluded that maintaining a temperature of 68 F at the 30 in. level is a more acceptable test standard than maintaining

TABLE 5—SUMMARY OF RESULTS OF TESTS OF FRONT OUTLET CONVECTORS AND BASEBOARDS

	TEST NUMBER								
	11	12	13	14	15	16	17	18	19
Air Temperatures, F:									
Center of room:									
3 in. above floor	66.3	62.5	64.7	63.0	63.9	61.3	61.1	64.4	59.8
6 in. above floor	66.9	63.7	65.9	64.2	65.1	63.0	62.7	67.0	62.2
12 in. above floor	68.1	65.2	67.6	65.7	66.9	64.5	64.2	68.1	63.2
18 in. above floor	68.7	67.0	69.0	67.1	68.8	66.1	66.3	69.5	64.6
24 in. above floor	70.4	68.3	70.6	68.5	70.2	67.2	67.1	70.2	65.3
30 in. above floor	73.2	69.4	72.5	70.0	71.4	67.7	67.4	70.5	65.6
36 in. above floor	75.7	70.3	73.9	71.6	72.5	68.2	68.1	70.8	66.0
48 in. above floor	80.5	72.7	76.3	74.8	74.2	69.8	70.6	71.6	67.2
60 in. above floor	83.8	74.8	78.9	76.6	77.1	72.0	72.0	72.6	68.2
72 in. above floor	84.3	75.2	79.7	76.6	77.3	71.8	71.9	72.4	67.5
93½ in. above floor	88.5	79.2	82.8	80.6	80.7	76.3	76.0	73.4	68.9
12 in. from west wall:									
3 in. above floor	65.8	62.0	64.3	62.4	63.4	60.6	60.6	64.8	60.2
6 in. above floor	66.6	64.2	65.6	64.6	64.8	62.1	63.1	66.3	61.7
12 in. above floor	67.1	65.0	67.2	65.2	66.7	63.9	63.9	67.5	63.1
18 in. above floor	68.4	66.1	68.6	66.4	68.2	65.1	65.3	68.5	63.9
24 in. above floor	70.1	67.8	70.3	67.8	69.9	66.1	66.3	69.5	64.8
30 in. above floor	72.7	68.7	71.9	69.4	71.1	66.6	67.0	69.4	65.2
36 in. above floor	75.9	70.6	73.7	72.1	73.0	68.4	68.6	71.0	66.2
48 in. above floor	80.1	72.5	76.6	74.8	75.3	70.0	70.2	71.6	67.1
60 in. above floor	82.1	73.7	78.6	76.3	76.2	70.7	71.6	72.4	68.1
72 in. above floor	85.7	75.6	80.8	78.3	78.0	72.4	72.8	73.2	69.0
93½ in. above floor	90.0	78.6	83.4	79.6	80.2	75.0	74.9	75.3	71.2
Air inlet to units[a]:									
18 in. in front of unit	67.3	63.5	65.7	63.5	64.4	61.8	61.8	65.5	61.6
36 in. in front of unit	66.5	63.0	65.3	63.2	64.1	61.1	60.8	65.0	60.6
Cold space near cold walls, F	14	14	14	14	14	14	14	14	14
Globe Thermometer (room center):									
12 in. above floor	68.2	65.3	67.7	65.6	66.8	64.0	63.9	67.6	62.9
48 in. above floor	77.4	70.4	74.2	72.2	72.7	68.1	68.3	70.2	65.4
Total rate of condensation, lb/hr	7.16	6.39	6.82	6.44	6.52	6.11	6.14	6.50	6.14
Net rate of condensation, lb/hr	7.05	6.19	6.62	6.24	6.32	5.91	5.94	6.50	6.14
Net rate of heat emission, Btu/hr	6827	5994	6411	6043	6120	5723	5752	6295	5946

[a] At one-half height of air inlet for convectors and 3 in. above floor for baseboard.

70 F at the 5 ft level. In discussing this paper, S. R. Lewis pointed out the need for a new term to describe the numerical magnitude of effective heat.

In 1931, Trane and Scanlan[8] reported the results of tests on cast iron radiators and convection heaters in a room with three water-cooled steel walls and with no infiltration; the tests showed that the rate of heat loss from a given room is dependent upon the type of heating unit used and that heating effect or effective heat measures this change in the overall rate of heat transfer per degree of temperature difference.

In 1932, Kratz and Fahnestock[9] compared the results obtained from testing four types of convectors with an unenclosed 5-tube cast iron radiator when a temperature of 68 F at a level of 30 in. above the floor was maintained in all

TABLE 6—SUMMARY OF TESTS OF SLANT TOP AND TOP OUTLET CONVECTORS AND NON-STANDARD RADIATORS

	Test Number											
	21	22	23	24	31	32	33	34	35	36	41	42
Air Temperatures, F:												
Center of room:												
3 in. above floor	66.3	66.4	63.9	69.7	61.8	63.0	59.2	61.3	73.5	77.9	61.7	62.5
6 in. above floor	66.7	67.8	65.1	72.2	62.0	64.1	59.9	62.1	75.4	80.0	62.5	63.5
12 in. above floor	68.0	69.6	66.7	74.0	63.3	65.9	61.4	63.9	77.4	82.4	63.8	64.7
18 in. above floor	68.6	71.5	68.4	76.2	63.7	67.3	62.9	65.3	79.3	84.5	64.4	65.6
24 in. above floor	70.0	72.9	69.9	77.2	64.8	68.9	64.4	66.7	80.9	85.8	65.6	66.3
30 in. above floor	72.3	74.5	71.5	77.9	66.4	70.7	65.8	68.4	82.2	87.2	67.0	67.6
36 in. above floor	75.1	76.4	72.8	79.1	68.5	71.8	66.9	69.7	83.0	88.5	68.4	69.0
48 in. above floor	79.9	79.4	75.6	81.5	73.6	74.7	69.8	72.1	85.2	90.7	72.0	71.7
60 in. above floor	84.6	82.5	78.0	83.8	78.0	77.7	71.7	75.3	88.9	94.5	75.7	74.2
72 in. above floor	85.2	82.3	78.0	83.4	79.2	78.0	71.6	75.9	88.2	94.5	76.8	75.7
93½ in. above floor	89.9	86.0	81.4	88.2	84.3	82.6	75.1	79.5	94.5	101.6	81.5	80.1
12 in. from west wall:												
3 in. above floor	65.7	65.6	63.3	69.1	61.2	62.0	58.4	60.4	72.5	77.1	61.4	61.9
6 in. above floor	66.5	67.0	65.4	71.3	61.9	63.3	59.9	61.3	74.7	79.0	61.9	62.8
12 in. above floor	66.9	69.2	66.6	73.4	62.2	65.2	61.4	63.0	76.8	81.6	62.5	63.9
18 in. above floor	68.2	70.7	67.9	75.1	63.4	66.6	62.5	64.4	78.2	83.4	63.5	64.8
24 in. above floor	69.6	72.5	69.6	76.4	64.3	68.6	64.0	66.3	80.0	85.0	65.4	65.9
30 in. above floor	72.0	74.0	71.1	77.6	66.3	70.0	65.4	67.5	81.3	86.8	66.9	67.4
36 in. above floor	75.3	76.1	73.2	79.2	68.9	72.0	67.2	69.7	83.1	88.5	68.7	69.0
48 in. above floor	80.1	78.9	75.7	81.2	74.0	74.4	69.5	72.1	85.5	91.0	72.5	71.6
60 in. above floor	82.5	81.6	77.1	82.2	76.8	76.5	71.0	74.1	87.0	92.9	74.7	73.1
72 in. above floor	85.6	84.9	79.3	84.6	79.3	78.8	72.8	76.8	89.8	95.4	77.1	76.2
93½ in. above floor	91.5	85.7	81.0	86.9	85.7	82.3	74.9	79.2	93.3	99.2	82.9	79.4
Air inlet to units[a]:												
18 in. in front of unit	67.0	66.7	63.9	69.9	63.1	63.7	59.3	61.9	74.4	78.4	63.0	64.4
36 in. in front of unit	66.4	66.5	64.1	69.6	62.5	63.6	59.2	61.8	73.6	78.1	61.7	63.6
Cold space near cold walls, F	14	14	14	14	14	14	14	14	14	14	14	14
Globe Thermometer (room center):												
12 in. above floor	68.3	69.3	66.8	73.4	63.6	65.8	61.4	63.8	77.0	82.0	63.9	65.0
48 in. above floor	77.1	76.7	73.1	79.2	71.1	72.4	67.4	70.4	83.1	88.4	69.9	70.0
Total rate of condensation, lb/hr	7.18	7.18	6.51	7.48	6.51	6.68	5.83	5.94	8.02	8.50	6.50	6.46
Net rate of condensation, lb/hr	7.07	6.98	6.31	7.28	6.40	6.48	5.63	5.74	7.82	8.30	6.39	6.26
Net rate of heat emission, Btu/hr	6847	6759	6111	7050	6198	6275	5452	5559	7573	8038	6188	6062

[a] At one-half height of air inlet for convectors and 2½ in. above floor for radiation.

cases. The heights of the convector enclosures were between 26 in. and 30⅜ in., while the height of the cast iron radiator was 26 in. The output of the radiator was adjusted by wrapping sections, while the outputs of the convectors were adjusted by blocking off part of the enclosures and heating elements. They concluded that the convectors condensed from 1.5 to 7.1 percent less steam than the unenclosed cast iron radiator when maintaining 68 F at the 30-in. level.

In 1940, Kratz, Fahnestock, Broderick, and Sachs[10] reported on tests of radiators and convectors. Three-tube and five-tube cast iron radiators with a height of 26-in. and 3-tube radiators with a height of 25 in. were tested, and convector enclosures with a height of 26 in., 28 in., and 29 in. were tested. In

TABLE 7—VALUES OF EFFECTIVE HEAT FOR FRONT OUTLET CONVECTORS AND BASEBOARDS

HEIGHT OF INDEX POINT, IN.	q	TEST NUMBER								
		11	12	13	14	15	16	17	18	19
		6827	5994	6411	6043	6120	5723	5752	6295	5946
3	Δt	52.3	48.5	50.7	49.0	49.9	47.3	47.1	50.4	45.8
	q_n	7020	6415	6760	6490	6625	6240	6215	6710	6040
	EH	2.8	7.0	5.4	7.4	8.2	9.0	8.0	6.6	1.6
6	Δt	52.9	49.7	51.9	50.2	51.1	49.0	48.7	53.0	48.2
	q_n	6990	6500	6830	6570	6710	6400	6350	7010	6280
	EH	2.4	8.4	6.5	8.7	9.6	11.8	10.4	11.4	5.6
12	Δt	54.1	51.2	53.6	51.7	52.9	50.5	50.2	54.1	49.2
	q_n	6960	6530	6880	6605	6780	6435	6395	6960	6260
	EH	2.0	8.9	7.3	9.3	10.8	12.4	11.2	10.6	5.3
18	Δt	54.7	53.0	55.0	53.1	54.8	52.1	52.3	55.5	50.6
	q_n	7010	6760	7060	6775	7025	6635	6660	7140	6420
	EH	2.7	12.8	10.1	12.1	14.8	15.9	15.8	13.4	8.0
24	Δt	56.4	54.3	56.6	54.5	56.2	53.2	53.1	56.2	51.3
	q_n	7085	6795	7120	6825	7055	6655	6640	7060	6400
	EH	3.8	13.4	11.1	12.9	15.3	16.3	15.4	12.2	7.6
30	Δt	59.2	55.4	58.5	56.0	57.4	53.7	53.4	56.5	51.6
	q_n	7270	6760	7175	6840	7025	6540	6500	6900	6280
	EH	6.5	12.8	11.9	13.2	14.8	14.3	13.0	9.6	5.6
36	Δt	61.7	56.3	59.9	57.6	58.5	54.2	54.1	56.8	52.0
	q_n	7375	6670	7140	6840	6955	6420	6405	6740	6160
	EH	8.0	11.3	11.4	13.2	13.6	12.2	11.4	7.1	3.6
48	Δt	66.5	58.7	62.3	60.8	60.2	55.8	56.6	57.6	53.2
	q_n	7410	6460	6885	6700	6630	6145	6230	6340	5880
	EH	8.5	7.8	7.4	10.9	8.3	7.4	8.3	0.7	−1.1
60	Δt	69.8	60.8	64.9	62.6	63.1	58.0	58.0	58.6	54.2
	q_n	7260	6210	6670	6405	6465	5910	5910	5970	5510
	EH	6.3	3.6	4.0	6.0	5.6	3.3	2.8	−5.2	−7.3
72	Δt	70.3	61.2	65.7	62.6	63.3	57.6	57.9	58.4	53.5
	q_n	7020	6010	6500	6160	6240	5630	5660	5710	5210
	EH	2.8	0.3	1.4	1.9	2.0	−1.6	−1.6	−9.3	−12.4
93½	Δt	74.5	65.2	68.8	66.6	66.7	62.3	62.0	59.4	54.9
	q_n	6860	5965	6295	6090	6100	5710	5680	5480	5090
	EH	0.5	−0.5	−1.8	0.8	−0.3	−0.2	−1.2	−13.0	−14.4
Average EH for all heights		4.2	7.8	6.8	8.8	9.3	9.2	8.5	4.0	0.2

these tests, equivalent temperatures at the 30 in. level were measured by a thermo-integrator. Some of the authors' conclusions were: the use of a test room as large as 15 ft by 18 ft with an 8 ft 6 in. ceiling height tends to reduce inherent differences in performance of radiators and convectors but does not alter these

TABLE 8—VALUES OF EFFECTIVE HEAT FOR SLANT TOP AND TOP OUTLET CONVECTORS AND NON-STANDARD RADIATORS

HEIGHT OF INDEX POINT, IN.	q	TEST NUMBER											
		21	22	23	24	31	32	33	34	35	36	41	42
		6847	6759	6111	7050	6198	6275	5452	5559	7573	8038	6188	6062
3	Δt	52.3	52.4	49.9	55.7	47.8	49.0	45.2	47.3	59.5	63.9	47.7	48.5
	q_n	7020	7040	6625	7590	6315	6490	5950	6240	8260	9060	6300	6415
	EH	2.5	4.2	8.4	7.7	1.9	3.4	9.1	12.2	9.1	12.7	1.8	5.8
6	Δt	52.7	53.8	51.1	58.2	48.0	50.1	45.9	48.1	61.4	66.0	48.5	49.5
	q_n	6960	7140	6710	7880	6255	6560	5970	6270	8450	9290	6320	6470
	EH	1.6	5.6	9.8	11.8	0.9	4.5	9.5	12.8	11.6	15.6	2.1	6.7
12	Δt	54.0	55.6	52.7	60.0	49.3	51.9	47.4	49.9	63.4	68.4	49.8	50.7
	q_n	6945	7190	6750	7910	6270	6640	6020	6355	8480	9360	6340	6460
	EH	1.4	6.4	10.5	12.2	1.2	5.8	10.4	14.3	12.0	16.4	2.5	6.6
18	Δt	54.6	57.5	54.4	62.2	49.7	53.3	48.9	51.3	65.3	70.5	50.4	51.6
	q_n	7000	7450	6965	8210	6300	6800	6190	6520	8730	9680	6390	6560
	EH	2.2	10.2	14.0	16.5	1.6	8.4	13.5	17.3	15.3	20.4	3.3	8.2
24	Δt	56.0	58.9	55.9	63.2	50.8	54.9	50.4	52.7	66.9	71.8	51.6	52.3
	q_n	7030	7450	7015	8130	6340	6880	6285	6590	8730	9560	6440	6530
	EH	2.7	10.2	14.8	15.3	2.3	9.6	15.3	18.6	15.3	18.9	4.1	7.7
30	Δt	58.3	60.5	57.5	63.9	52.4	56.7	51.8	54.4	68.2	73.2	53.0	53.6
	q_n	7150	7460	7040	7960	6375	6930	6300	6630	8610	9410	6450	6530
	EH	4.4	10.4	15.2	12.9	2.9	10.4	15.6	19.3	13.7	17.1	4.2	7.7
36	Δt	61.1	62.4	58.8	65.1	54.5	57.8	52.9	55.7	69.0	74.5	54.4	55.0
	q_n	7295	7470	6990	7840	6450	6860	6260	6600	8390	9200	6440	6510
	EH	6.5	10.5	14.4	11.2	4.1	9.3	14.8	18.7	10.8	14.5	4.1	7.4
48	Δt	65.9	65.4	61.6	67.5	59.6	60.7	55.8	58.1	71.2	76.7	58.0	57.7
	q_n	7330	7270	6800	7540	6560	6680	6145	6395	8020	8750	6380	6350
	EH	7.0	7.6	11.3	6.9	5.8	6.5	12.7	15.0	5.9	8.9	3.1	4.8
60	Δt	70.6	68.5	64.0	69.8	64.0	63.7	57.7	61.3	74.9	80.5	61.7	60.2
	q_n	7360	7100	6570	7260	6570	6540	5875	6260	7910	8640	6310	6140
	EH	7.5	5.0	7.5	3.0	6.0	4.2	7.8	12.6	4.5	7.5	2.0	1.3
72	Δt	71.2	68.3	64.0	69.4	65.2	64.0	57.6	61.9	74.2	80.5	62.8	61.7
	q_n	7125	6800	6315	6920	6445	6315	5630	6090	7500	8320	6180	6060
	EH	4.1	0.1	3.3	−1.8	4.0	0.6	3.3	9.6	−1.0	3.5	−0.1	0.0
93½	Δt	75.9	72.0	67.4	74.2	70.3	68.6	61.1	65.5	80.5	87.6	67.5	66.1
	q_n	7015	6600	6165	6830	6435	6275	5605	5990	7540	8460	6180	6050
	EH	2.4	−2.4	0.9	−3.1	3.8	0.0	2.8	7.8	−0.4	5.2	−0.1	−0.2
Average EH for all heights		3.9	6.2	10.0	8.4	3.1	5.7	10.4	14.4	8.8	12.8	2.4	5.1

characteristics sufficiently to affect materially conclusions drawn from similar tests in smaller rooms; radiators and convectors that produce the least air temperature differences between ceiling and floor give the smallest rates of steam condensation while maintaining 68 F at the 30 in. level; the equivalent tem-

peratures produced by radiators are somewhat higher than those produced by convectors for the same air temperature. In the large room, three different convectors, each with an enclosure height of 26 in. showed, as an average, a condensation of 0.1337 lb of steam per hr for each degree of temperature difference between air at the 30 in. level in the center of the test room and air in the cold space, while the condensation rate of the 3-tube radiator 25 in. high

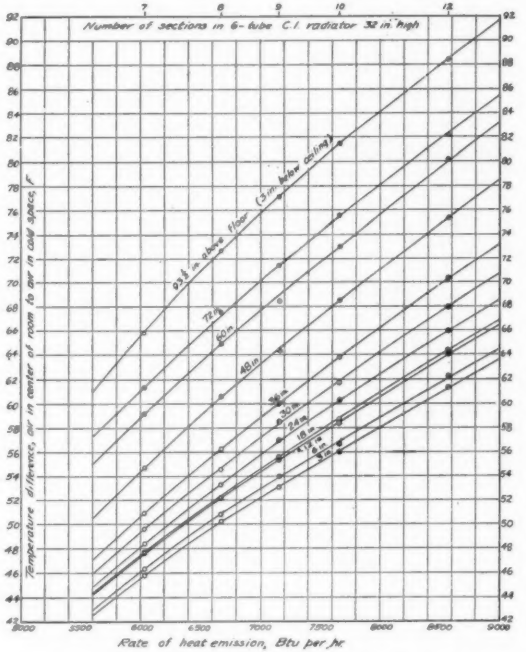
Number of sections in 6-tube C.I. radiator 32 in. high
Temperature difference, air in center of room to air in cold space, F
Rate of heat emission, Btu per hr

FIG. 7. PERFORMANCE OF STANDARD CAST IRON RADIATORS

and the 5-tube radiator 26 in. high was 0.143 lb per hr for each degree of temperature difference so measured. These results were obtained with an air temperature in the center of the room at the 30 in. level of 68.0 F ±0.2 deg, and a temperature of the cold space of −1.3 F ±0.3 deg, and indicate an average effective heat for the 26 in. enclosure of 6.4 percent when compared with the radiators.

In the Commercial Standard CS 140-47 of the United States Department of Commerce for Testing and Rating Convectors, there appears a table for additions to observed ratings of convectors with front outlets (Table 1).

For ratings of convectors with top outlets, no additions to observed capacity are allowed.

For ratings of convectors with inclined outlets, the maximum additions to capacity are found by multiplying the tabular values for front outlets by the ratio of the angle of outlet with the horizontal to 90 deg. For example, a 30 deg slant top convector with the top of the outlet 24 in. above the floor would have a maximum addition to observed capacity of (30/90) (9) or 3 percent.

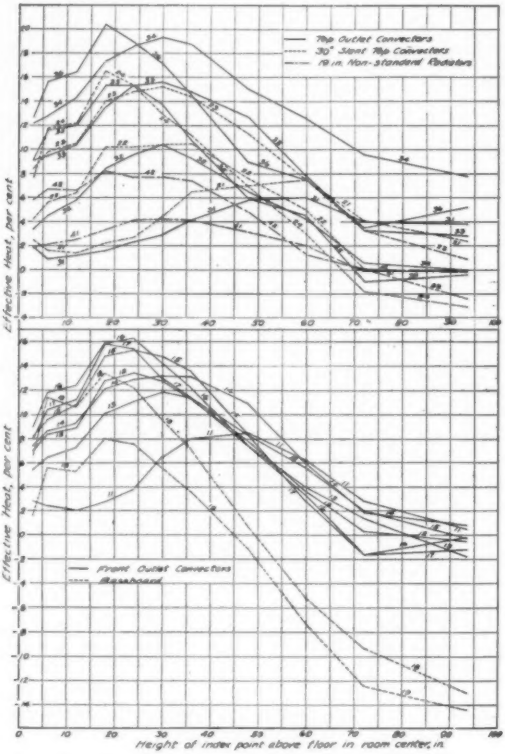
Top Outlet Convectors
30° Slant Top Convectors
19 in. Non-standard Radiators
Effective Heat, per cent
Front Outlet Convectors
Baseboard
Height of index point above floor in room center, in.

FIG. 8. EFFECTIVE HEAT AT DIFFERENT LEVELS OF THE INDEX POINT

PURPOSE OF TESTS

The purpose of the test program suggested by the Engineering Committee of the *CMA*, consisting of E. G. Powell, R. D. Tutt, and E. L. Broderick (consultant), was to determine the condensate necessary to heat a structure to a given condition when using devices of different outlet air ejection heights and locations and to determine a new standard against which to compare the performance of the convectors.

TEST PROCEDURE

All radiators, convectors, and baseboards were tested in the cold room of the Sibley School of Mechanical Engineering at Cornell University with a procedure that complied with Commercial Standard CS 140-47. The construction and arrangement of this cold room are shown in Fig. 1. The cold room is enclosed in a refrigerated shell space that is 26 ft × 29 ft with a ceiling height of 11 ft 11 in. Cold air is supplied through a horizontal duct that runs above the inner cold or test room; air is recirculated to the coil room at the floor level of the shell space. As indicated in Fig. 1, the west and north walls of the test room simulate exposed walls, while the south and east walls, floor, and ceiling are

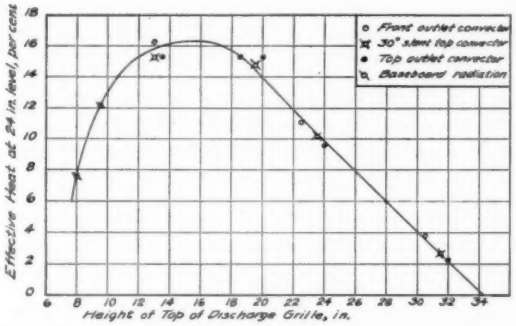
Effective Heat at 24 in. level, per cent
Height of Top of Discharge Grille, in.
Front outlet convector
30" slant top convector
Top outlet convector
Baseboard radiation

FIG. 9. EFFECTIVE HEAT OF DIFFERENT HEATING UNITS AT THE 24-IN. LEVEL

more heavily insulated. Calculated values of the overall coefficients of heat transfer for the structure appear in Table 2.

During most of the tests reported, air was supplied through a continuous slot around the periphery of the north window at a controlled rate of 12 cfm by a small blower taking cold air from the shell space (see Fig. 2). In this way, infiltration of air through window cracks, amounting to one-half air change per hour was simulated. This is equivalent to an additional heat loss of 13 Btu per (hr) (F deg), making the total for the room 112.5 Btu per (hr) (F deg). In all tests, the air temperature in the cold shell space was held at 14 F ±½ deg. Air motion in the test room was a result of natural circulation, and the relative humidity in the test room was well under 75 percent in all tests.

Steam was supplied to each unit tested at a controlled pressure of 31.75 in. Hg absolute, which is the saturation pressure at 215 F; the degree of superheat in all tests was 5 F. Condensate was collected outside of the test room in a conventional collector with an air vent.

Air temperatures were measured by copper-constantan thermocouples located at the center of the room and also in a vertical line 12 in. from the west wall at the following elevations above the surface of the floor: 3 in., 6, 12, 18, 24, 30, 36, 48, 60, 72, and 93½ in. (3 in. below the ceiling) (Fig. 3). Also globe thermometers with a diameter of 2 in. were used in the center of the room at eleva-

tions of 12 in. and 48 in. In the tests of all radiators, the air inlet temperatures were measured with shielded copper-constantan thermocouples at a height of 2½ in. above the floor and at distances of 18 in. and 36 in. in front of the center of the radiator face. (Fig. 4) In the tests of all convectors, the air inlet temperatures were measured with shielded copper-constantan thermocouples located in accordance with Section 19d, CS 140-47, at one-half the height of the inlet air grille and 18 in. in front of the convector inlet. Also, a second air inlet temperature was observed 36 in. in front of the convector inlet (Fig. 3). A high top outlet convector and a low front outlet convector under test are shown in Figs. 5 and 6.

The cast iron radiators used as the standards were 6 tubes in depth and 32 in. high. These were located as shown in Fig. 4 on the north wall between the window and the west wall. Five sizes were used with 7, 8, 9, 10, and 12 sections, with no wrapping or blocking of any sections. Each radiator was painted with a flat prime coat of medium gray oil paint and was located 1½ in. from the wall and had a total pitch of ¼ in.

The convector enclosures had top, back, and sides of 20-gage steel and fronts of 18-gage steel. The inside and outside of the enclosures were painted with a flat prime coat of medium gray oil paint. The air inlet was without grille and the air outlet was a typical deflection type grille. All units with an overall height of 24 in. or less were located under the window on the north wall, while those with an overall height of more than 24 in. were located on the north wall between the window and the west wall. A total pitch of ¼ in. was given each unit, and a small insulated copper tube vented the element continuously outside of the test room.

The convector heating elements were constructed of ½ in. O.D. copper tubes with aluminum fins. The fin thickness was 0.013 in., with 6 fins per inch, and the fin height was 2 in. on all elements. The length of each element was ⅜ in. less than the overall length of the enclosure.

The dimensions of the heating units are shown in Table 3, and the test schedules and the observed test results are given in Tables 4 to 6, inclusive. The effective heat values at different index point heights for the various heating units are shown in Tables 7 and 8.

DISCUSSION OF RESULTS

From the tests of the cast iron radiators 32 in. high, which were used as standards, (Table 3) the curves of Fig. 7 were plotted. In this figure, the difference between the temperature of the air in the test room at a given index point and the temperature of the air in the cold shell space is plotted as ordinate *vs.* the net rate of heat emission of the radiator as abscissa. There are eleven curves, one for each height of the index point in the center of the room. In the test of a non-standard unit, the temperature difference at the index point (Δt) and the net rate of heat emission (q) are observed; the rate at which the standard radiator would emit heat (q_s) to maintain the same temperature differential is then read from Fig. 7. The effective heat in percent of the non-standard unit at the chosen index point is found from the relation

$$EH = 100\ (q_s/q - 1)$$

For example, (Table 5) in test 15 of the front outlet convector 20 in. high, Δt at the 24 in. level is $70.2 - 14 = 56.2$ F, and $q = 6120$ Btu per hour. In Fig. 7 the standard cast iron radiator would have to emit heat at the rate of $q_s = 7055$ Btu per hour to maintain a differential of 56.2 F at this level. The percent effective heat is $100\ (7055/6120 - 1)$ or 15.3.

CONCLUSIONS

1. For each unit tested, there is one level of the index point that gives maximum effective heat. This is clearly shown in Fig. 8. For convector enclosures with the same type of air discharge, the elevation of this point increases as the height of the enclosure increases. For enclosures of the same height but with different types of air

TABLE 9—COMPARISON OF PERCENT OF EFFECTIVE HEAT WITH HEIGHT OF INDEX POINT

HEIGHT OF TOP OF ENCLOSURE ABOVE FLOOR IN.	OUTLET TYPE	MAXIMUM EFFECTIVE HEAT, PERCENT	HEIGHT OF INDEX POINT FOR MAXIMUM EFFECTIVE HEAT, IN.
13½	front	16.3	24
	30° slant	16.5	18
	top	15.3	18, 24
20	front	15.3	24
	30° slant	15.2	30
	top	15.6	30
24	front	11.9	30
	30° slant	10.5	36
	top	10.4	30
32	front	8.5	48
	30° slant	7.5	60
	top	6.0	60

discharge, the height of the index point for maximum effective heat increases as the air is given a greater upward deflection at the outlet.

The practical significance of this result is that the height of front outlet enclosures should not exceed 20 in., while slant and top outlet convectors should be from 3 in. to 6 in. lower if maximum effective heat is desired at a low level, 24 in. A collection of some of the results in Table 9 illustrates this conclusion.

2. At very high and very low levels in the center of the test room, all units tested showed low values of effective heat. For all heights of front outlet enclosures, the effective heat 3 in. above the floor was between 2.8 and 9.0 percent; for the baseboards at this level, the effective heat was between 1.6 and 6.6 percent. For 30 deg slant top convectors, the values were between 2.5 and 8.4 percent; for top outlet convectors, between 1.9 and 9.1 percent.

At a point 3 in. below the ceiling, the front outlet convectors showed effective heats between −1.8 and 0.5 percent; for baseboards, these effective heats were between −14.4 and −13.0 percent. For 30 deg slant top convectors, these effective heats were between −3.1 and 2.4 percent; for top outlet convectors, between −0.4 and 3.8 percent.

3. Based upon the results of these tests, alone, a suggested table of effective heats as shown in Table 10.

The principal difference between this table and the one given in Commercial Standard CS140-47 is in the performance of the slant top and top outlet convectors. For a

given height above the floor of the top of the discharge grille, these tests showed no significant difference in the effective heats at the 24-in. level of different types of outlets.

In Fig. 9 are shown effective heats at the 24-in. level plotted against the height of the top of the discharge grille above the floor.

4. The effect of infiltration on effective heat of the same heating unit is not positively established because the number of tests were insufficient. However, tests 33 and 34 may be compared to show the effects of infiltration upon the same top outlet convector with an enclosure height of 20 in.; in test 33, the infiltration was at the controlled rate of 12 cfm, while in test 34, the air slots around the window admitted no air. With infiltration, the maximum effective heat was 15.6 percent, and the average effective heat at all levels was 10.4 percent; without infiltration, the maximum effective heat was 19.3 percent, and the average effective heat at all levels was 14.4 percent. It must be remembered that the performance of the convector, both with

TABLE 10—SUGGESTED EFFECTIVE HEAT VALUES FOR VARIOUS HEIGHTS OF DISCHARGE GRILLE

HEIGHT OF TOP OF DISCHARGE GRILLE, IN.	EFFECTIVE HEAT (24 IN. LEVEL) FOR FRONT, SLANT TOP, OR TOP OUTLETS, PERCENT
8	7
10	13
12	15
14	16
18	15
20	14
21	13
22	12
23	11
24	10
25	9
26	8
27	7
28	6
29	5
30	4
31	3
32	2
33	1

and without infiltration, is compared with the performance of the standard radiator *with* infiltration in finding these effective heats.

It is a fact that infiltration should cause a lower room air temperature and a greater rate of heat emission for a given temperature of the outdoor air *if* infiltration does not change the output of the heating unit per degree of temperature difference between heating fluid and room air. Without infiltration (test 34), the average temperature of the air in the center of the test room at all levels was 69.1 F, and the rate of heat emission was 5559 Btu per hour; with controlled infiltration (test 33), this average temperature was 66.2 F, and the rate of heat emission was 5452 Btu per hour. If infiltration of cold air above the convector decreases the rate of heat emission per degree of temperature difference between heating fluid and room air, it would be possible to get lower room air temperatures with about the same, or even lower, rates of heat emission. Apparently, there is a pronounced effect of infiltration upon this rate of heat emission. Similar results were obtained when a top outlet convector with an enclosure height of 13½ in. was tested with and without infiltration.

5. The 12 section, 6-tube, non-standard cast iron radiator with a height of 19 in. showed a maximum effective heat of 4.2 percent at a height of the index point of

30 in. when located in the same place as the standard radiator (test 41). When the 19-in. radiator was placed under the window, a maximum effective heat of 8.2 percent was obtained at a height of the index point of 18 in. (test 42). These results indicate that when the standard heating unit is placed at only one location in the test room, while the non-standard unit is tested in several different locations, different values of effective heat may be found for the same heating unit in the same room.

6. These tests show conclusively that the same structure with the same temperature of the outdoor air, the same average temperature of the air (all levels) *in the center of the structure*, the same controlled infiltration, and the same heating fluid (steam) may have different rates of heat loss with different heating units.

For the five standard cast iron radiators, the rate of heat loss from the structure has an average value of 118.9 Btu per hour for each degree of difference between the average room air temperature and the outdoor air temperature. For twelve convectors of different types and heights the corresponding average rate of heat loss is 111.7 Btu per (hr) (F deg). This difference shows the influence of the local or direct effect of the type of heating unit on the rate of heat loss. The temperature of the room surface of the wall back of the 9-section standard radiator in test 3 was found to be 146.2 F, while the temperature of the surface of the wall back of the front outlet convector in test 16 was found to be 101.3 F.

If, instead of averaging the air temperature at all levels in the center of the room in order to find the heat loss per degree of temperature difference, *one* air temperature at the 24 in. level in the room center is used, the average rate of heat loss for the five standard radiators is 128.4 Btu per (hr) (F deg), while for the twelve convectors the corresponding rate of heat loss is 116.9 Btu per (hr) (F deg). Comparison of these results shows the influence of temperature change with level within the room as well as the influence of local or direct effect of the heating unit on the heat loss.

The temperature variation with level and the local heat loss near the heating unit are characteristic of a particular type of heating unit, and both affect the rate of heat loss from a given structure.

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DISCUSSION

W. S. HARRIS, Urbana, Ill. (WRITTEN): Effective heat is a subject that has received a great deal of attention during the past twenty years. There never has been unanimity of opinion on the subject, but the data presented by the authors of this paper are a valuable contribution to the literature and should help in reaching common ground for agreement.

First, there are three questions I would like to ask the authors, and then I wish to discuss certain phases of the results.

1. Was there any insulation in the wall in back of the baseboard units? The way that these units are installed makes them a part of the wall and unless adequate insulation is provided in back of them the heat loss through the wall may be abnormally high. I believe most manufacturers recommend filling the stud spaces with insulation to a minimum height of 12 inches when baseboard units are installed in uninsulated houses. Their recommendation should be followed in laboratory set-ups as well.

2. Were any tests made with the walls of the cold room insulated? The test data show that the effective heat values are closely related to floor to ceiling temperature differences. This temperature difference, in turn is largely dependent upon the construction of the walls of the room. I am wondering if perhaps effective heat is not more a function of house construction than of type of heating unit?

3. From Fig. 2 it appears that the infiltration air is discharged into the stud space well below the window. Would this not reduce the wall surface temperature below that normally expected? If so, it could have an adverse effect on the performance of some heating units, especially of those which transmit heat to the room through the wall as well as directly to the room air.

Effective heat factors as calculated in this paper are arbitrary values in that they are based on the performance of an arbitrarily selected *standard* heating unit operating under arbitrarily selected conditions. These selections should represent typical usage in so far as it is practical to do so, if the term effective heat is to have practical value. Reliable statistics on factory shipments of cast iron radiators during the past five years indicate that the most common height of radiator is 25 in. These statistics indicate that over 60 percent of all cast-iron radiators sold since 1945 by those companies reporting was of this height, whereas 32-in. radiation represented only 8 percent of the total volume. In view of this it would seem that the 25-in. 4-tube or the 25-in. 5-tube radiator would be a more logical choice for a standard unit than the less frequently used 32-in. 6-tube unit. Also the standard unit should be located under the window to be consistent with recommended installation practice and since the majority of the other units were so located.

There is no question in my mind but that the method of test has made it possible to determine the effective heat values for the units included in the scope of the tests when operated in accordance with the prescribed conditions in a cold room. I do question whether or not these values will apply when the units are used under normal operating conditions. I also question the value which should be placed on effective heat based on dry bulb temperatures alone, neglecting entirely any radiation effects. In support of these statements the following discussion is presented.

Fig. A is a plot of the effective heat values based on 24 in. level air temperature as given in Tables 7 and 8 of the paper *vs.* the floor to ceiling air temperature difference as determined from data in Tables 5 and 6. Tests 34 and 36 have been omitted from Fig. A as they represent a different operating condition (no infiltration) than do the other tests. Except for the two baseboard tests, there is a distinct correlation between the floor to ceiling temperature difference and the value of effective heat. The answer to the question concerning the method of installing the

baseboard may throw some light on the reasons why the baseboard units did not follow this same pattern.

In test No. 42, using a 19-in. radiator under a window, the floor to ceiling temperature difference was 17.6 deg, while the same height radiator operating on steam in the Research Home produced a floor to ceiling temperature difference of only 6.2 deg. In view of the apparent relationship between floor to ceiling temperature differences and effective heat I think it is important that the cause of the wide spread between floor to ceiling temperature differences obtained in the cold room and in a residence operated under actual use conditions be explained.

Since we are more interested in producing a comfortable environment than in producing a given dry bulb temperature, it is logical that radiation effects should be included in the determination of effective heat. About 1933 a series of tests somewhat similar to those reported in this paper were made in the cold room at

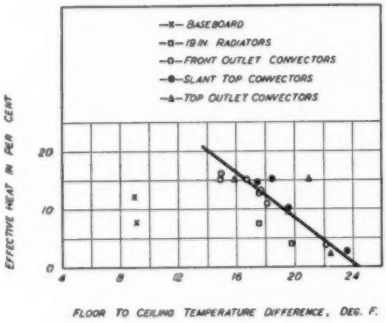

FIG. A. EFFECTIVE HEAT BASED ON DRY BULB TEMPERATURE

the University of Illinois. In the tests at Illinois the heat requirements for the different units were compared both for a fixed dry bulb temperature near the center of the room and 30 in. above the floor, and for a fixed equivalent temperature at this same location as determined by a eupatheoscope. A 26-in., 5-tube radiator was used as the standard. These tests indicated that a 24-in. front outlet convector had a heating effect of about 13 percent based on dry bulb temperature alone, but based on equivalent temperature, the heating effect of this same unit was only about 7 percent. While the tests at Illinois agree very well with the Cornell tests when only the dry bulb temperature is considered in determining effective heat values, the Illinois tests further demonstrated that when radiation effects were considered by basing effective heat on equivalent temperature the actual effective heat values were reduced by almost 50 percent, thus indicating the importance of considering radiation as well as convection when attempting to compare the performance of different types of heating units.

The definition of effective heat states that heating units having a high effective heat value will heat a room to a given temperature at a fixed point with a smaller number of heat units than will be required by a heating unit with a low effective heat value, and that the amount of reduction in heat required is in direct proportion to the effective heat value. In other words, a unit having an effective heat value of 15 percent will require 15 percent less heat to heat a room to a given

temperature at a fixed point than will be required by a heating unit having an effective heat value of 0. If the effective heat values apply under actual use conditions as well as the prescribed test conditions the fuel consumption required to heat a given house with different types of heating units would be in inverse proportion to the effective heat values of the different units used.

During the ten years that tests of the performance of steam and hot water systems have been made in the Research Home at the University of Illinois, this house has been heated using radiators, convectors, and baseboard units. Fig. B presents the daily heat input to the I=B=R Research Home over a range of indoor-outdoor temperature differences as obtained when using front outlet convectors with the top of the outlet grille 22 inches above the floor, 19-in., 4-tube radiators, and type RC radiant baseboard. The same boiler, burner, and general piping arrangement was used in each case. The fuel burning rate and the CO_2 content of the flue gas was

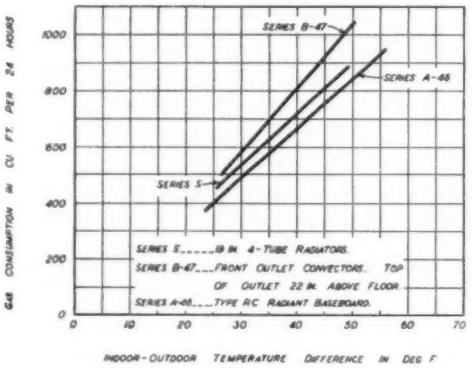

FIG. B. DAILY FUEL CONSUMPTION CURVES

held constant. Hot water was the heating medium and the design water temperature was 215 F. From data in the paper it appears that the value of effective heat for the baseboard might be as high as 12 percent, for the 19-in. radiator 8 percent, and 12 percent is suggested for the convector. The fuel consumptions in Fig. B obviously do not follow this same pattern. At an indoor-outdoor temperature difference of 34 deg, typical of average winter conditions at Urbana, Illinois; the average daily fuel consumption when using the baseboard was 8.4 percent less than when using the 19-in. radiators while the average daily fuel consumption when using the convectors was 11.7 percent higher than that obtained with the radiators. Thus while the baseboard and the convector have about the same effective heat values as defined in the paper there was some 20 percent difference in actual daily fuel requirements with these two types of heating units.

The data just given leads me to believe that there is a vast difference in operating characteristics of any heating unit when operated at steady state conditions in a cold room as compared to intermittent operation encountered in an actual installation. The great need at this time is further study of the causes of the differences between laboratory and field results so as to make it possible to translate laboratory results into values suitable for use at operating conditions met in actual practice.

It also leads me to believe that the thing which has been defined as effective heat is really a usage factor which is more dependent upon the structure, the method of

installation, and the method of operation than on the design of the heating unit itself and therefore should not be based on unit design.

To summarize my remarks:

1. A 25-in. radiator would be a more representative standard than the 32-in. radiator. The former represents over 60 percent of the volume of radiator shipments while the latter represents about 10 percent.

2. The standard unit should be located under the window so as to be consistent with recommended installation practice.

3. The method of test as proposed by the authors provides data which can be used to calculate an effective heat factor for a unit operating in a given room under steady state conditions.

4. Radiation effects in an enclosed heated space cannot be overlooked in the determination of any effective heat factor.

5. If effective heat factors are to be allowed for one part of the industry, then an adequate and acceptable procedure for determining effective heat must be developed which will apply to all types of heating units and effective heat factors should be applied to all units demonstrating that such a factor is warranted.

6. No evidence is presented which would indicate the degree to which room construction would influence the test results. This information is needed before it is safe to say differences in test results obtained in different test rooms would be inconsequential.

7. In suggesting a table of effective heat values it is assumed that the authors are suggesting these for use in establishing catalog ratings. This tacitly implies that the values for effective heat as determined by conditions of steady state operation apply equally well to the non-steady state conditions of operation encountered in actual house heating practices. The validity of this implication is questioned for the following reasons:

a. Data contained in the paper indicate that there is good correlation between effective heat and the floor to ceiling temperature difference. The floor to ceiling temperature differences in the cold room were about three times as large as those observed in the Research Home with the system operating under normal use conditions.

b. The definition of effective heat states that for given temperature conditions the heat loss of the room is a function of the effective heat value. Therefore, if the values of effective heat obtained by the cold room tests also apply for actual use conditions the fuel consumption required to heat a given space would be affected by the effective heat value pertaining to the unit being used as a heat source. Fuel consumption data for hot water heating systems in the Research Home do not substantiate this argument.

8. This paper is a valuable addition to the literature on the subject of effective heat. It suggests an experimental technique and a method for computing effective heat factors.

It gives the results of a large number of tests conducted with a number of representative heating units and demonstrates that the magnitude of the effective heat factors depend upon the structure, the unit selected as standard, the location of the standard and non-standard units, the level in the test room used as the index point, the amount of the infiltration, etc. In the case of one type of radiator, one type of convector and one type of baseboard, the results presented in the paper are not in agreement with those obtained with these same units when used under actual service conditions in the Research Home. This emphasizes the need for additional work toward developing a laboratory procedure which will give practical results applicable to actual field use.

R. S. DILL, Washington, D. C. (WRITTEN): The purpose of rating convectors or other heating equipment is not to compare one device to another. The chief fact

which I wish to establish is that the purpose of rating heating devices is to permit selection of them so that their capacities fit the requirements of the rooms or spaces they are to serve. Use of oversized heating equipment is traditional, presumably because oversizing results in less subsequent trouble than undersizing and because accurate heat loss computations are difficult and troublesome. This is the reason why convector ratings including a so-called heating effect factor have been with us for some years. At the present time the accent is on economy of metal and this will promote precision in heat loss computations including the raising of design outdoor temperatures in many localities and more accurate estimates of the infiltration loss for houses or buildings. When heat loss calculations are really accurate, use of heating effect factors will become intolerable because it will result in the choice of equipment with too little capacity.

The heating effect concept is objectionable in that it is illogical. It ascribes a rating to a device which is greater than its observed capacity. On the contrary, computation of the heat loss of a building by THE GUIDE's recommended method is a logical process and its only imperfections, if they can be called such, are due to such unknowns as the infiltration loss, but the formula for heat loss involving areas, temperature differences and a law of heat transmission is based on logic. Accurately applied, the formula would indicate a heat supply for a room or structure, and it would be impossible to heat the room or structure with a smaller supply. Use of a heating effect factor is based on the false premise that a room can be heated with less heat than it requires.

In a conventional room temperature stratification is usual, and it can result in a heating requirement exceeding by 15 percent or more that which would be required if uniform temperature distribution were attained. Hence, research into the heatability of rooms or spaces may be worthwhile. The uncertainty in this subject is indicated on pp. 233 and 234 of THE GUIDE 1950 where some suggestions are given for more accurate computation of heat loss based on assumed temperature conditions. In all cases an increase in the capacity of the heating device selected is implied which is directly opposed to the result when heating effect factors are used. In the warm air industry this effect is also recognized, and the FHA now recommends a 10 percent factor of safety when warm air supply grilles are placed near the ceiling. This is virtually the application of a negative heating effect. The truth is that insofar as a heating device is responsible for heat loss due to bad temperature distribution, it is inefficient and it could properly be given an efficiency, less than unity, instead of a credit obtained by comparison with another device which is considered inferior.

For many years it has been recognized that a better heat distribution and a better economy could be obtained with a low rather than a high heating device, and the chief reason for this is the greater heat loss through the ceiling and upper portions of the walls with a bad temperature distribution. This fact has been used in attempts to justify heating effect factors for convectors. A salesman on one occasion, however, was frank enough to admit that convectors needed heating effect factors in order to face competition. This being so, it is apparent that heating effect factors should be set by the sales department and not on engineering considerations in any given company.

Heating effect factors are used in connection with Commercial Standard ratings simply because I was heavily outvoted in Committee. Commercial Standards are based to a large extent on the democratic principle of majority rule and, as chairman of the Rating Committee, I have approved the ratings of many convectors which included the prescribed allowance for heating effect. The only achievement attributable to my efforts is the requirement in the Commercial Standard that heating effect factors used in computing ratings shall be included in manufacturers' catalogs or other tabulated rating data. This is proper since it informs the engineer or designer so that his computations can be suitably amended if need be. So long as the facts

are disclosed, particularly to Government engineers or designers, my essential function has been performed.

A series of tests essentially similar to that described in the present paper has frequently been discussed at the National Bureau of Standards and elsewhere in the past, and the usual conclusion was that such a project was practically futile for several reasons, and I am not convinced to the contrary now. The first reason is that the results can be qualitatively predicted; another is given on page 236 in the paper where it states, *It must also be clear that standard and non-standard heating units when tested in test room A may give different values of effective heat than when both are tested in test room B.* In other words, the results depend on the conditions of tests, in particular on the characteristics of the test room. In this case the rooms' characteristics were palpably fortunate for the sponsors of the project, since high heating effects are indicated for convectors of the more popular sizes. The low heating effect indicated for an 8-in. device must bring grief to manufacturers of floorline radiators or convectors, and perhaps they will see fit to expend some thousands of dollars to demonstrate that such devices have compensating virtues.

The authors state that the characteristics of the test room can also affect the results of capacity tests, and this is admitted, but the effect is demonstrated to be minor since both convectors and radiators have been tested in cold wall rooms, cool wall rooms, and open booths, all specified in the Commercial Standard, and the results agree with acceptable accuracy.

Assignment of the same heating effect to top outlet, slant outlet, and front outlet convectors is tantamount to saying that a room is equally well heated with any of the three types, and this is contrary to common experience. Also, and again, the inferior performance of the low device, with the top of its discharge grille 8 in. above the floor is mysterious and contrary to test data obtained at the National Bureau of Standards and elsewhere. The only theory that accounts for the poorer performance of the low device is that there was no insulation, or very little insulation, on the back of it.

Comparison of devices some of which are located under a window and others of which are located adjacent to the wall beside the window and then attributing observed differences in performances to the characteristics of the devices is not regarded as justifiable. The ventilating slot around the window used in these tests is certainly not a conventional thing, and its use may account in part for the anomalous performance reported for the low heating devices. It is noted that the radiator selected for a standard is 32 in. high instead of 38 in. used by some investigators heretofore and that the so-called index point is 24 in. above the floor, which is lower than anyone has attempted to standardize before, so far as I know. There is nothing peculiarly beneficial about the 32-in. radiator any more than there is for the 38 in. A 24 in. for the index or reference point is regarded as too low since it is obviously below the center of gravity of a person even when seated.

To summarize in conclusion, I believe it should be established that the purpose of ratings for heating systems or devices is to permit selection of capacities to fit requirements; research into the heatability of rooms is desirable; and no heating devices should have a rating greater than their observed capacity.

R. H. WEIGEL* and H. W. WILLIAMS,* Johnstown, Pa. (WRITTEN): The authors have defined effective heat in terms of the ratio of the heat emission of a *standard* heating unit to the heat emission of a *non-standard* heating unit required to maintain the same air temperature at an index point in the test room as the *standard* with the same external conditions. The selection of the standard heating unit, hence, assumes a role of great importance and should be selected with great care. First, it should be representative of the wet-heat industry as determined by popular usage and

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secondly, it should be installed with the recommended practice of the day. The selection of the 32-in., 6-tube radiator as a *standard* by the authors fulfills neither of the above requirements in that (1) it could not be installed under the window and (2) I=B=R statistics show approximately 60 percent of all cast iron radiators sold since 1945 to be of the 25 in. height and only 5 to 11 percent to be of the 32 in. height during this same period.

The importance of the selection and location of the *standard* heating unit is again emphasized when the data gathered by the authors is examined. In Test 41, twelve sections of 19-in., 6-tube cast-iron radiation were installed between the window and the west wall of the test room (as was the *standard* radiator), while in Test 42 the 19-in. radiator was installed under the window. At the index point of 24 in. above the floor in the center of the room, the 19-in. radiator produced an effective heat of 4.1 percent when at the same location as the *standard* radiator. When moved

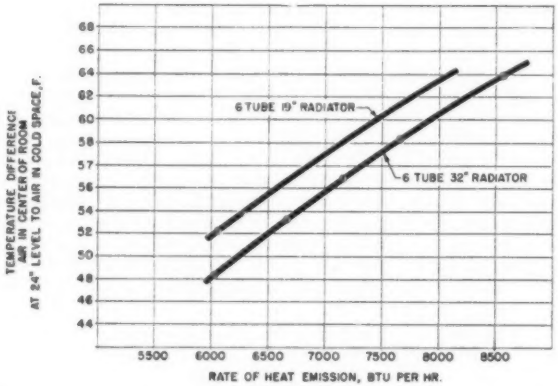

FIG. C. PERFORMANCE CURVE FOR 6-TUBE, 19 IN. RADIATOR AS COMPARED TO CURVE FOR 6-TUBE, 32 IN. RADIATOR

under the window the effective heat of the 19-in. radiator increased to 7.7 percent. In the latter case the 19-in. radiator required 485 Btu less heat emission than the 32-in. radiator to maintain a temperature difference of 53.2 F between the index point and the air in the cold space. If this reduction in heat emission can be assumed to be the same for temperature differences up to 66 F, a performance curve for the 19-in. radiator can be drawn as shown in Fig. C. Using the lower heat emissions required of the 19-in. radiator reduces the effective heats of the other units tested to the values shown in Table A. These values plotted against the height of the top of the discharge grille define the lower curve shown in Fig. D and suggests the effective heat values shown in the second column of Table B. It is to be noted that if the 19-in. radiator was used as a *standard* rather than the 32 in., no effective heat percentages would be added to convectors having the top of their discharge grille over 27 in. while for heights of discharge grilles lower than this, the minimum reduction in effective heats would amount to 40 percent. It is realized that the validity of these percentages are questionable due to the assumption made in determining the performance curve of the 19-in. radiator, but they do emphasize the need for more experimental work before any definite conclusions are reached as to effective heat.

Up to the present point the discussion has been based entirely on what effect changes in the location and selection of a *standard* radiator might have on effective

TABLE A—EFFECTIVE HEAT VALUES FOR DIFFERENT HEATING UNITS BASED ON THE 19 IN. CAST IRON RADIATOR AS STANDARD AND AIR TEMPERATURES 24 IN. ABOVE THE FLOOR AS THE INDEX POINT

	Enclosure Type								
	F	F	F	F	F	F	F	Type RC Baseboard	Finned Tube Baseboard
Enclosure Height from Floor	32	24	24	24	20	13½	12	10	8½
Height of Top of Discharge Grille	30½	22½	22½	22½	18½	13	11½	9½	8
Test Number	11	12	13	14	15	16	17	18	19
q Btuh	6827	5994	6411	6043	6120	5723	5752	6295	5946
Temp. Diff. Between Air at Index Point and Air in Cold Space, Δt	56.4	54.3	56.6	54.5	56.2	53.2	53.1	56.2	51.3
q_s Btuh	6600	6310	6635	6340	6570	6170	6155	6575	5915
% E. H. 19 in. as Standard	−3.3	5.3	3.7	4.9	7.4	8.0	7.0	4.3	−0.5
% E. H. 32 in. as Standard	3.8	13.4	11.1	12.9	15.3	16.3	15.4	12.2	7.6
Reduction in % E. H. Using 19 in. as Standard	7.1	8.1	7.4	8.0	7.9	8.3	8.4	7.9	8.1

	Enclosure Type							
	S	S	S	S	Top Outlet	Top Outlet	Top Outlet	Top Outlet
Enclosure Height from Floor	32	24	20	13	32	24	20	13½
Height of Top of Discharge Grille	31½	23½	19½	13½	32	24	20	13½
Test Number	21	22	23	24	31	32	33	35
q Btuh	6847	6759	6111	7050	6198	6275	5452	7573
Temp. Diff. Between Air at Index Point and Air in Cold Space, Δt	56.0	58.9	55.9	63.2	50.8	54.9	50.4	66.9
q_s Btuh	6545	6965	6530	7645	5855	6275	5452	8245
% E. H. 19 in. as Standard	−4.4	3.1	6.9	8.4	−5.5	1.9	6.4	8.9
% E. H. 32 in. as Standard	2.7	10.2	14.8	15.3	2.3	9.6	15.3	15.3
Reduction in % E. H. Using 19 in. as Standard	7.1	7.1	7.9	6.9	7.8	7.7	8.9	6.4

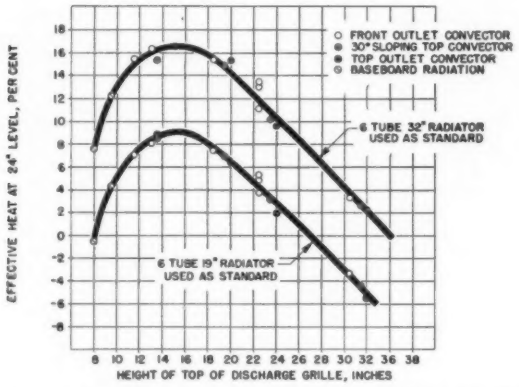
FRONT OUTLET CONVECTOR
30° SLOPING TOP CONVECTOR
TOP OUTLET CONVECTOR
BASEBOARD RADIATION
6 TUBE 32" RADIATOR USED AS STANDARD
6 TUBE 19" RADIATOR USED AS STANDARD
EFFECTIVE HEAT AT 24" LEVEL, PER CENT
HEIGHT OF TOP OF DISCHARGE GRILLE, INCHES

FIG. D. COMPARISON OF EFFECTIVE HEATS AT 24 IN. LEVEL—TWO RADIATORS USED AS STANDARD (A) 6-TUBE, 19 IN., AND (B) 6-TUBE, 32 IN.

heat values of various heating units if the effective heat values are based on the heat emission required to maintain the same air temperature at an index point of 24 in. above the center of the floor of the test room in a low temperature testing plant and with the air temperature surrounding the test room being held constant.

TABLE B—COMPARISON OF EFFECTIVE HEAT VALUES

HEIGHT OF TOP OF DISCHARGE GRILLE FROM FLOOR, INCHES	EFFECTIVE HEAT (24 IN. LEVEL) FOR FRONT, SLANT TOP, AND TOP OUTLET CONVECTORS		EFFECTIVE HEAT CS140-47		
	19 In. Radiator as Standard	32 In. Radiator as Standard	Front Outlet	30° Slant Outlet	Top Outlet
8	0	7			
10	5	13			
12	8	15			
14	9	16			
16	9	15			
18	8	14	15	5	0
20	7	13	13	4⅓	0
21	6	12	12	4	0
22	5	11	11	3⅔	0
23	4	10	10	3⅓	0
24	3	9	9	3	0
25	2	8	8	2⅔	0
26	1	7	7	2⅓	0
27	0	6	6	2	0
28		5	5	1⅔	0
29		4	4	1⅓	0
30		3	3	1	0
31		2			
32		1	2	⅔	0
33		0			
34			1	⅓	0
35					
36			0	0	0

The use of dry bulb temperatures alone at the index point as a basis for determining effective heat values does not take into account any radiation effects from surrounding surfaces. Would not this penalize heating units such as the radiant baseboard that warms the entire surface of the portion of the exposed wall along which they are installed as well as units presenting a larger area of direct radiating surface than the standard?

Long low heating units are at a disadvantage insofar as steam condensation is concerned if the wall against which they are placed is not insulated. Units of this type cover larger wall areas than do other heating units of comparable output and, consequently, it is desirable to reduce the heat transmission coefficient, U, of the wall in back of the unit to a minimum so that heat loss through this section of wall is not excessive. All of the tests reported in the paper were for uninsulated exposed walls. Would the same effective heats have been obtained if the exposed walls had been insulated?

The air temperatures obtained at the index point, and upon which the effective heats were based, were the result of a continuous rate of heat emission from the unit being tested. If the steam had been supplied intermittently as encountered in actual practice would the test units have had the same operating characteristics as when steam was supplied continuously?

The questions raised in this discussion are not to be construed as criticism of the fine contribution of Mackey, et al, on a controversial subject as effective heat. Rather it is to indicate the need for additional experimentation and research on such a debatable subject.

M. W. McRAE, Chicago, Ill. (WRITTEN): The data presented in this paper by Professor Mackey and his colleagues show some interesting relationships between the performance of various sizes of convectors and one height and depth of cast iron radiators. However, the results obtained do not seem to establish the validity of heating effect as a measurable quantity any more than did earlier work along similar lines.

The term heating effect or effective heat is not defined in Commercial Standard CS140-47, largely because the members of the Joint Convector Code Committee, who wrote the Standard, were unable to formulate a mutually satisfactory definition. Neither has THE GUIDE listed this term in the definition section, possibly for a similar reason.

The percentages given in Table 3 of CS140-47 are arbitrary values representing a compromise between various schools of thought within the committee and advisory group. The suggested effective heat factors listed in the table under conclusion 3 of this paper agree, within 1 percent, with those in the Commercial Standard. Such close correlation is remarkable.

Years ago, when earlier attempts were made to evaluate heating effect, the standard of comparison was the old, 3 column, large tube 38-in. high radiator. The *standard* used in the present investigation is the 6-tube, 32-in. high radiator of the current slim tube design. Neither of these radiators could be classified as outstanding examples of the ideal heating device as it is well known that high, wide, relatively short radiators produce much larger floor-to-ceiling temperature gradients than do long, low, narrow radiators.

The following questions are presented for consideration of the authors:

1. Why should a cast iron radiator of any type be used as a standard of comparison in determining the performance and rating of Convectors?
2. If a low, narrow radiator had been used as a standard, what effect would this have on the effective heat percentages listed under Conclusion No. 3?
3. Would it not be equally logical to determine heating effect for radiators by comparing their performance with that of a convector having a 32-in. high top outlet enclosure?

4. If a comparison of the performance of radiators and convectors is to be made, why not compare units having at least approximately the same over-all dimensions? Thus, the effect of the shape factor would be minimized and the heat outputs would represent the difference in performance attributable to the basic design of radiators *vs.* convectors.

It is felt that until a more basic standard of comparison can be found the measurement and evaluation of heating effect cannot be definitely established.

P. R. ACHENBACH, Washington, D. C. (WRITTEN): The test method used by these investigators appears to be well-planned for obtaining comparative information on radiators and convectors in that a more or less typical room was available and outdoor temperatures and infiltration could be controlled at the same level for all specimens. There appear to be some reasons, however, to question the suitability of the choice of some of the test conditions and test equipment in the following details:

1. A 32-in. 6 tube cast-iron radiator is probably not a representative standard for comparison with all convectors.
2. A valid comparison is not obtained in my opinion when the standard radiator is located beside the window and the convector is placed beneath the window.
3. The infiltration provided around the window affects the performance of units beneath it differently from those beside the window.
4. An air-temperature thermocouple at the center of the room is not necessarily the best device available for comparing two types of heating units one of which delivers almost all of its heat by convection and the other of which emits nearly half of its heat by radiation. An air-temperature thermocouple would favor the convection device. Comfort depends on mean radiant temperature as well as air temperature. Therefore, a globe thermometer or some other instrument sensitive to both radiation and convection would be a better index instrument.

The tabulated results tend to support these conclusions in several of the tests.

(a) The 32-in. radiator and the 32-in. convectors differ very little in effective heat (an average of 2.9 percent with the 24 in. level index). This difference may be almost entirely accounted for by differences in reverse heat flow through the wall back of the radiator.

The 19-in. radiators showed a higher effective heat than the 32-in. convectors. A comparison of tests 41 and 42 shows that the location beneath the window increased the effective heat for the 19-in. radiator. It would probably do the same for other radiators.

(b) Tests 33, 34, 35, 36 show that infiltration reduced the temperatures near the ceiling more than it did at the lower levels (including the 24-in. level) for the same convector located beneath the window. For devices located below the window the infiltration air mixes with the rising stream of warm air above the test specimen and produces lower ceiling temperatures, whereas for devices located beside the window the cold air proceeds toward the floor and mixes with room air before approaching the radiator or convector. In the latter case the rising current of warm air proceeds toward the ceiling without dilution causing higher ceiling temperatures and greater heat losses. Thus, the two locations are not comparable.

(c) The position beneath the window would be more favorable to the radiator using an index point at the center of the room because the index would be directly in front of the radiator. This is revealed in comparisons of Tests 41 and 42 and Tests 18 and 19.

The reduction of heat output of a given convector when infiltration was provided around the window suggests that the downward current of cold air obstructed or interfered with the upward current of warm air to the extent that the total mass flow of air through the convector was diminished.

The concept of effective heat as defined in this paper should not be applied to the levels above the living zone (say 5 or 6 ft above the floor). A positive value of

effective heat for a particular device represents a desirable characteristic. However, a negative effective heat, as defined in this paper, is desirable in the space above the living zone because overheating of the upper part of a house is generally undesirable. Clarification of this usage is especially important in the case of the baseboard units which show large negative values of effective heat between the 60 in. level and the ceiling.

It is believed that the comparisons between radiators and convectors would have to be broadened considerably before general conclusions about effective heat could be stated. A graph such as Fig. 9 in this paper should be developed using a 4 or 5 tube radiator from 19 to 23 in. high placed beneath the window for comparison with convectors located beneath the window. Comparisons based on an index temperature obtained with a globe thermometer would have a greater significance than those based on air temperature only.

M. K. FAHNESTOCK, Urbana, Illinois (WRITTEN): Persons working with the development of testing and rating codes for radiators, convectors, and radiant baseboards have been familiar with the fact that many factors affect the heat emission rates of these various heating units. However, it has been proven that when all of the influencing factors are properly taken into account, similar heat outputs will be obtained for various sizes of any one type of heating unit, in widely different test rooms or spaces. This is most important because insofar as possible, no laboratory rating should be contingent upon the physical test set-up, or widely different from the rating or performance which will be obtained in most heating installations.

The test facilities and procedures given in Commercial Standard CS140-47, Testing and Rating Convectors, mentioned in the paper are based upon results and experiences obtained in a number of industrial, university, and government laboratories. It should be noted that under certain conditions of test, instrumentation, and test procedure, that this commercial standard permits capacity or heat output tests to be conducted with convectors, in three widely different test rooms or spaces, namely, the warm-wall booth, the cooled-wall room, and the cold room. Within reasonable limits convectors may be tested in any of these set-ups with the same result insofar as *heat output* is concerned. However, it should be emphasized that this Commercial Standard applies only to the Testing and Rating of Convectors and not to the testing and rating of radiators and baseboards. In the paper the authors have reported on tests conducted with radiators, baseboards and convectors in a test room which conforms to the cold room specified in the commercial standard covering only the testing and rating of convectors.

It might also be mentioned that while the commercial standard includes a table of arbitrarily selected percentages which may be added to heat capacities as determined by test, in order to arrive at the ratings of certain types and heights of convectors, it does not provide for measuring or determining them. The authors have advanced a test procedure and method of analysis to establish these percentages. In the paper they are referred to as effective heats.

In the paper results are given on tests made with five 32-in. high, 6-tube free standing radiators of different lengths; 17 convectors, ranging in height from 12-in. to 32-in., two types of baseboards and one 19-in., 6-tube free standing radiator. All of the 32-in. high radiators were tested alongside the window in the exposed north wall and adjoining the exposed west wall; the 32-in. high convectors were also tested alongside of the window; fourteen of the convectors, ranging from 12-in. to 24-in. high, were tested under the window; and the 19-in. high radiator was tested both under the window and between the window and the west wall.

Unless I am mistaken, the 19-in. high radiator was the only unit tested in the two positions. Using the index point at the 24-in. level from Table 8, the effective heat factor for this radiator when located under the window is given as 7.7, and when located between the window and the west wall in the same position as the stand-

ard radiator the factor is reduced 4.1, a reduction of 47 percent. This agrees with conclusion 5 given in the paper, which shows that the effective heat factor may be varied by merely changing the position of the heating unit.

Referring to Table 5, it may be noted that practically the same 24-in. level index point temperature existed during test No. 19 with the finned-tube baseboard (65.3 F) and during test No. 41 with the 19-in., 6-tube radiator located alongside the window (65.6 F). Thus, as a matter of interest, the effective heat factors of these two units can be compared, taking one or the other as the standard. If the 19-in., 6-tube radiator located alongside the window is used as a standard with the 24-in. level as the index point, the finned-tube baseboard as reported in test No. 19 has an effective heat factor of 4.1 percent. In Table 7, Test 19, when the effective heat factor for the finned-tube baseboard is computed using the 32-in. 6-tube radiator located alongside the window as a standard, the effective heat factor is given as 7.6 percent. This is an increase of 85.5 percent and it illustrates how the value of the effective heat factor is dependent on whatever unit is taken as the standard.

It is my understanding that the authors arbitrarily selected the 32-in. 6-tube free-standing radiator as the standard heating device and located it alongside the window in the north exposed wall and adjacent to the west exposed wall. Since fourteen of the seventeen convectors tested were tested under the window, it would seem that it would have been more logical to have selected as a standard a type of radiator which could have been tested under the window in the same position as the convectors. Another reason for locating both the standard and non-standard heating devices in similar positions under the window is that this location is well recognized in practice.

In conclusion 6 it is stated that under the prevailing test conditions, 118.9 Btu per hour was the average rate of heat loss from the structure per degree temperature difference between the average room air and the outdoor air when the room was heated with the 32-in., 6-tube, radiators located alongside the window exposed north wall and also that for the twelve convectors of different types and heights and presumably located under the window in the exposed north wall and away from the exposed west wall, the corresponding average rate of heat loss was 111.7 Btu per hour. No mention is made of the corresponding values for the 19-in., 6-tube radiator located under the window or for the two types of baseboards. These values calculated from data given in Tables 4 and 5 are 110.0 Btu for the 19 in., 6-tube radiator located under the window; 112.2 Btu for the Type RC baseboard and 115.8 Btu for the finned-tube baseboard. Thus, on the basis of the heat loss per degree temperature difference between the average indoor air and the outdoor air the heat loss from the structure is a minimum when the 19-in., 6-tube radiator was used under the window.

The results given in the paper are interesting and they demonstrate that many factors may affect the performance of various types of heating units. The type and height of the heating unit selected for a standard, its location relative to the exposed window and the non-standard units and the selection of the 24-in. level index point for indoor temperatures used in the analysis of the results, while convenient for the purposes of the study, do not, in my opinion, conform sufficiently with current practice. I would like to see the work continued using a small-tube radiator, either 19 in. or 25 in. high and located under the exposed window in the same position as the non-standard units and an index point at the 30 in. level in the center of the room adopted.

I would also like to see the performance of the various heating units compared on some evaluation of the comfort conditions produced in the test room, rather than upon the heat loss from an arbitrarily selected test structure. While heat losses are important, the end objective for heating an enclosed space occupied by human beings is to maintain a comfortable ambient environment for the human occupants. It is well known and recognized that in normal rooms heated with radiators, baseboards and

convectors, the comfort of the occupants from the standpoint of thermal balance depends largely upon the environmental factors of dry-bulb temperature and mean radiant temperature. Therefore, in a non-uniform environment, neither the temperature of the indoor air, the heat loss from the structure, or a combination of them, is an index of the effectiveness of heating units designed for heating spaces for human occupancy. While they are not entirely satisfactory, it is my opinion that for conditions and purposes of tests corresponding to those reported in this paper, observations made with instruments such as the globe thermometer, the eupatheoscope, and the thermo-integrator will give more valuable relative data on the effectiveness of different heating units in producing environments conducive to human comfort. All of these instruments respond to both the air temperature, and the radiation present in an enclosed space.

R. N. TRANE, LaCrosse, Wis. (WRITTEN): The authors of the paper "Principles of Effective Heat for Steam-Heated Structures" are to be commended on the thorough analysis they have given the general subject of effective heat. Those of us who have been in the industry for many years realize that this is a subject that has been controversial from the day it was first considered. Although some research work has been done on the subject in the past, it has failed to settle the controversy.

The approach used by Professor Mackey and his associates is a new one. From the results obtained it is possible to analyze the relative performance of various types of heating equipment and to separate, at least in part, factors contributing to heating effect. The test procedure is also unique in a manner in which standard performance is achieved without blanketing, wrapping or otherwise interfering with the normal function of the heating equipment.

The selection of the heating unit to be used as a standard may cause some controversy. However, I believe that in the last analysis this particular issue is not too important. If we are to evaluate the performance of various types and shapes of heating equipment some standard must be employed. The cast iron radiator selected is a standard product of accepted rating. The installation is one commonly encountered. It would be accepted engineering practice to utilize such a radiator installed in such a manner to offset a calculated heat loss equivalent to the catalog output of the radiator. It should, therefore, be a type of unit and type of application which should satisfy the requirements of a standard device.

These are the points which I found of particular interest in analyzing the results obtained from this report:

1. The average effective heat percentages obtained at the general 24 to 30 in. index level match quite closely the heat effect percentages sanctioned by Commercial Standard CS140-47 for front outlet convectors. It is interesting to note that Professor Mackey finds that top outlet and slant top outlet units have approximately the same heating effect percentages, although the Code penalizes these convector types in the addition of heating effect. In this connection there is a practical problem involved in the performance of top outlet units in severe climates where downdrafts from windows may act to reduce the output of a unit. The heating effect factors in the Code for top outlet units have attempted to take this fact into account.

2. The manner in which Professor Mackey and his associates have presented their data permits us to further analyze the data to determine the factors causing these high values of effective heat for under window units. Following the same procedure used by the authors in determining the effective heat percentages, an analysis was made to determine what portion was a result of floor to ceiling temperature gradients within the room for a specific unit. These results are shown in Fig. E.

It is evident from the curve that approximately 50 percent of the effective heat percentage is the result of an improvement in floor to ceiling temperature gradient. The remaining portion can only be attributed to a change in overall heat transfer coeffi-

cient and/or a reduction in effective temperature difference across the structure. It can be probably assumed that this decreased heat loss is in the vicinity above and in back of the unit.

3. It seems to me that the test results justify the practice established by the Commercial Standard CS140-47 of adding effective heat percentages to the condensate output of convectors. It is certainly a more rational method of taking into account the effective heat percentages than by figuring heat losses in the conventional manner and then reducing these losses according to the type of heating equipment that is to

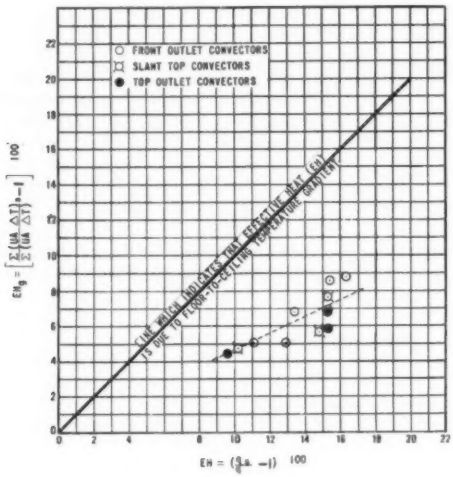

FIG. E. ANALYSIS OF THE PART FLOOR TO CEILING TEMPERATURE GRADIENTS PLAY IN CAUSING HIGH VALUES OF EFFECTIVE HEAT FOR UNDER WINDOW UNITS

EH_g = Portion of effective heat resulting from improved vertical temperature gradient in center of room with reference to 24 in. index point.
UA = Calculated values as listed in report (infiltration load included).
ΔT = (From the actual tests) Difference between temperature in center of room in each case minus cold space (plenum) temperature. (Weighted on basis of surface areas involved).
s — (subscript) — Standard unit which in this case was 6-tube 32 in. cast iron radiator.
All other symbols similar to those used in report. Based on data from report.

be installed. Many times, of course, the engineer does not know what type of equipment will be installed and is thus not able to exercise this judgment.

By calculating the actual heat loss of the structure, using the factors and the temperatures in the room as given by the author, the results as compared to the actual condensate output of the standard radiator, check within two or three percent. However, comparing the condensate rating of the convector against the heat loss calculated in the same manner, computations will show that appreciably less condensate is required to achieve the same temperature at the various index points. Unless proper recognition is given to this fundamental fact as established by the test results, the installation of unnecessarily over-sized heating equipment will follow.

4. My analysis brought out one more rather interesting fact concerning the baseboard convector. In the case of this type of heating unit, the heating effect as given in the report is apparently due solely to floor to ceiling temperature gradients.

5. The paper also brings out quite clearly the importance of under the window location of heating equipment, both from the standpoint of comfort and economy. This, of course, has been recommended procedure in the industry for many years. The paper supports standard practice.

It is my understanding that further research in this connection is to be conducted this year, utilizing hot water as the heating medium. I am looking forward with a great deal of interest to the results of this further study because of continuing growth in the use of hot water heating. In conclusion it appears to me that these are the factors that contribute to heating effect: Direction of convective air discharge, temperature gradients, location of the heating unit, the height of the heating unit, the extent of the infiltration, the type of heat output, *i.e.*, convection or radiation.

It appears that further research may establish the effect of these various factors on the total heating effect. Nevertheless, it seems to be well established that the effective heating percentages now established in Commercial Standard CS140-47 are conservative and in accordance with proper engineering procedure.

R. K. THULMAN, Washington, D. C.: *Effective Heat* or *Heating Effect* has plagued the industry almost from the inception of the idea. The reference to it in Commercial Standard, 140-47, and the inclusion at a table of allowances for it in the Standard was vigorously but unsuccessfully opposed by several of us on the Standing Committee. The test data in this paper seems to confirm the position we took.

I had hoped that this paper, prepared by such able investigators, would forever lay the ghost of this will-o-the-wisp. Unfortunately, it merely seems to provide an opportunity to extend the life of this unwarranted and misleading brainchild of sales promotion.

The conclusions and implications, etc., particularly the suggested table of allowances (which, incidentally, agrees remarkably with the guesses set forth in the Commercial Standard) do not appear to be justified by the data in the paper. Other choices of a standard, index point, location of the reference radiator, method of measuring temperature, design and construction at the test room, the fenestration at that room; choices other than those set forth in the paper and equally if not more representative of standard heating and building practices would have produced quite different results.

For example, had the reference radiator been installed under the window, a quite common practice, and had that radiator been lower and thinner than the window-sill, an equally common practice, results quite different from those reported in this paper would have produced, results based, for instance, on Test No. 42. Selection as a standard of the 19-in. radiator (Test No. 42), located under the window, would have resulted in an effective heat of only 4.5 percent, instead of 12.9 percent for the 24-in. convector with front outlet and located under the window. Had a three-tube radiator been used instead of a six-tube, or had there been another window in the room, this slight difference would probably have vanished.

In my humble opinion, the Society should not publish papers which might be construed by the public as having a potentially commercial character. I suggest that the present paper be returned to the investigators and resubmitted at a later date with conclusions more in line with a realistic consideration of more common heating practice.

R. W. MORGAN, Buffalo, N. Y.: Testing operations in our firm include not only convectors and other heat producing devices, but also cooling equipment. It would be helpful, particularly to our sales department, if the concept of effective cooling could be applied to room air conditioners. However, I doubt that there is justification for such a concept, any more than there is reason for the principle of effective heat.

I question whether the 24 in. level as used by the authors is the ideal height at which to measure comfort. As the authors indicated in this paper, there are so many possible variables in convector applications that I doubt the usefulness of the effective heat method of rating.

I think also we ought to look at the commercial side of this picture. If you analyze the great number of convectors that have been used as radiators in the past, I think you will find that they have done a fairly successful job in heating the buildings in which they were placed; and I don't believe we have had to add many convectors to bring sub-standard jobs up to capacity. This would indicate that our factor of safety in applying these devices must be a little bit on the high side.

It would seem to me that it would be better to raise our design temperatures ten deg and forget the effective heat factor. I think we would still be commercially successful and be a little bit more technically correct.

AUTHORS' CLOSURE (C. O. Mackey): The authors wish to thank all who have discussed this paper. The written discussion has been carefully prepared and adds materially to the value of this paper. The subject of effective heat has been a controversial one for some time, and it was not expected that these results, alone, would completely resolve the controversy.

Those who discussed this paper have repeated many of the same objections. Some of those objections will now be answered.

Harris, Williams and Weigel, Achenbach, and Fahnestock all question the use of dry-bulb temperature alone as an index of comfort to be used in calculating effective heat. Globe thermometers were placed at the 12 in. level and at the 48 in. level in the center of the test room. These readings are included in the test results as presented. If an operative temperature, which is the average of the dry-bulb temperature and mean radiant temperature, be used as an index, the convection coefficient of heat transfer for a globe thermometer with a diameter of two inches is such that the globe temperature is substantially the same as this operative temperature.

Effective heat based upon air temperature at the 48 in. level may then be compared with effective heat based upon globe temperature at the 48 in. level as indicated in Table C.

TABLE C—COMPARISON OF EFFECTIVE HEAT BASED ON AIR AND GLOBE TEMPERATURES AT 48 IN. LEVEL

UNIT	HEIGHT IN.	TEST NO.	EFFECTIVE HEAT AT 48-INCH LEVEL—PERCENT	
			BASED ON AIR TEMP.	BASED ON OPERATIVE TEMP.
Front outlet convector.....	32	11	8.5	6.6
Front outlet convector.....	13½	16	7.4	8.2
30° slant top convector.....	32	21	7.0	5.9
30° slant top convector.....	13½	24	6.9	6.5
Top outlet convector.....	32	31	5.8	5.2
Top outlet convector.....	13½	35	5.9	5.2
Baseboard.....	10	18	0.7	1.8
Baseboard.....	8½	19	—1.1	—1.3

From this table, it should be clear that operative temperature gives substantially the same results as dry-bulb temperature. The difference between a dry-bulb temperature and a temperature which includes a radiation factor, for test conditions similar to those reported in this paper, has been shown by a number of investigators to be small. There should be little doubt that dry-bulb temperature is the most reasonable and practical temperature to use.

Another common criticism questions the use of the 32-in. cast-iron radiator as the standard heating unit. It must be apparent that the selection of a standard unit, whether radiator, convector, or baseboard, is arbitrary. The test scheme was devised in such a way as to make this arbitrary selection as unimportant as possible.

The authors did not hesitate to give effective heat values for the 19-in. cast-iron radiator, and it was pointed out that the effective heat of this cast-iron radiator was greater when it was placed under the window than when placed at one side of the window. The selection of the 19-in. cast-iron radiator as the standard, as clearly shown by Williams and Weigel, would result in lower values of effective heat for the convectors. Such selection would also result in lower values of effective heat for the baseboard and negative values of effective heat for the 32-in. cast-iron radiator. Volume of radiator sales is not a scientific method of selecting the standard units.

The choice of a standard radiator is not as difficult a problem as the various discussors would suggest. Any radiator or convector could be used as a standard and, as such, a starting point from which to calculate effective heat percentages. The advantage of the standard used in the paper is the fact that so far, all of the usable effective heat percentages have been shown as positive values.

Probably the best example of the relative unimportance of a zero scale point is the Fahrenheit temperature scale where the zero is arbitrarily set and has no particular meaning, yet it does not cause difficulty.

Many of the written discussions mention the desirability of applying insulation in the wall in back of the baseboard. It is also desirable to apply insulation in the walls in back of any of the heating devices. It would be possible to apply insulation in back of each heating unit in capacity tests, but how much insulation should be used? The purpose of these capacity tests was to determine the rate of heat emission of the standard and non-standard units in the same structure. The authors feel sure that the results given in the paper would have been criticized very quickly if test conditions had been varied by using insulation for one test and not for another.

Intermittent supply of steam or other heating fluid may well result in a different characteristic of the heating unit than will appear in tests conducted under conditions of thermal equilibrium. It is customary, however, to conduct capacity tests of radiators, convectors, and baseboards under conditions of thermal equilibrium, and it might be well to emphasize again that these tests were all capacity tests, the results of which have been used to determine effective heat. Much more study than has yet been made in any one laboratory or research residence would be needed in order to define unsteady state characteristics because of the large number of variables entering this complex problem in heat transfer.

Mr. Dill's remarks are very interesting. Effective heat (or allowable additions to observed capacity) is certainly being tolerated in writing test codes. If numerical values are to be suggested and permitted in increasing observed capacities to give capacity ratings of convectors and baseboards, the authors of this paper believe that it is well that everyone understands where these values come from and what they mean. It might also be stated that Paragraph No. 52 of Commercial Standard CS140-47 for Testing and Rating Convectors seems quite clear in expressing the hope that a sound engineering basis may be found for effective heat values.

There are several statements in Mr. Dill's remarks to which we must take friendly exception. He states, *The heating effect concept is objectionable because it is illogical,* and that the only imperfections in the computation of the heat loss of a building are due to *such unknowns as the infiltration loss.* He also mentions certains *laws* of heat

transfer. A simple example will show how heat losses are commonly miscalculated from laws of heat transfer wrongly applied.

Consider an exterior wall twelve feet by eight feet with an overall coefficient of heat transfer of 0.15 Btu per (hr) (sq ft) (F deg). Assume the air temperature everywhere within the room is 70 F and that the outdoor air temperature is 0 F. Then, with steady heat flow the rate of transfer of heat through this wall is $96 \times (0.15) \times (70)$ or 1008 Btu per hr. The laws of heat transfer also say that there is no point in the wall structure where the local temperature is higher than 70 F.

Now, put a heating device directly in this room—radiator, baseboard, convector, or panel. If sufficient heat transfer surface be installed to give an air temperature of 70 F at a point 24 in. above the floor in the center of the room, how many of you believe that the air temperature is now 70 F. everywhere within the room? How many of you believe that there is still no point in the wall structure where the local temperature is higher than 70 F? How many of you believe that the rate of steady heat flow through the wall is still 1008 Btu per hr? How many of you think that the *laws* of heat transfer are properly applied in the conventional method of calculating heat loss?

This simple example illustrates the point that the authors have tried to make clear in this paper; namely, that the rate of heat loss will depend upon the heating unit installed. The example is directly opposed to Mr. Dill's statement that, *Use of a heating effect factor is based on the false premise that a room can be heated with less heat than it requires.* In the example, a direct relationship is shown between the heat distribution within a room and the heat loss from it. Rather than a consideration that heating effect is based on a false premise, there is evidence of another false premise that the ability of a heating device to condense steam is an accurate measure of its ability to distribute heat within a room.

Mr. Dill regards the 24-in. height of the index point as too low because it is below the center of gravity of a person even when seated. This might be a valid argument if it could be proved that the center of gravity is also the center of human sensitivity to physical environment. Dr. Brabbée, in some of his writings, recommended knee height as a good height of the index for comfort, while it is a well-known fact that the human ankle is extremely sensitive to temperature change.

Professor Harris questions the effect of the method of admitting the air that simulates infiltration. The wall panel to which this air is admitted is 4 ft wide; this panel is sealed from the rest of the north wall, and a large part of the panel is occupied by the window, so only a small portion of the wall is affected. In later tests, we propose to admit the air more directly to the slots and to determine any effect on the performance of the heating unit.

Professor Harris expresses some surprise over the floor-to-ceiling air temperature difference of 17.6 deg obtained with the 19-in. cast-iron radiator. This result is in line with results obtained in capacity tests in other laboratory test rooms. In tests at the University of Illinois,* floor-to-ceiling air temperature differences between 19.4 deg and 20.6 deg were reported for cast-iron radiators 25 in. and 26 in. high. These tests were conducted in a room 15 ft by 18 ft with an 8 ft 6 in. ceiling; the test room was then operated with two exposed walls, with the temperature of the air above the ceiling of the test room maintained at 62 F and with heating of the air under the floor.

Additional information was presented by Professor Harris in connection with Fig. B of his discussion, which purported to compare the fuel consumption of heating systems using baseboards, convectors, and radiators, and to show a relationship between effective heat and fuel consumption for a variety of indoor-outdoor temperature differences.

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* Effect of Room Dimensions on the Performance of Direct Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock, E. L. Broderick, and S. Sachs (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 331).

It should be noted that test points are not shown in the figure which was presented, and no supporting data are given to prove a basis of comparability for the curves.

It is evident that such data are made up of 24 hour totals which are subject to daily and seasonal weather conditions and are further involved because of the superimposed variables connected with the intermittent operation of a heating system. Such conditions are obviously different from the steady-state equilibrium conditions now used as a basis for rating this type of heating equipment.

Professor Fahnestock calculated the rate of heat loss from the structure for the 19-in. cast-iron radiator placed under the window (test No. 42) to be 110.0 Btu per hr per degree difference between average air temperature at room center and outdoor air. The *total* rate of heat emission and the rate of loss of heat from the structure during this test was 6256 Btu per hr; the average temperature of the air at room center was 69.2 F; the temperature difference between room air and cold space air

TABLE D—COMPARISON OF ACTUAL NET RATE OF HEAT EMISSION WITH CONVENTIONALLY CALCULATED HEAT LOSS

Test No.	Heating Unit	Actual Net Rate of Heat Emission Btu/hr q	Air Temp. at 24 In. Level F t_i	Calculated Heat Loss Btu/hr $q_i = 112.5\,(t_i\text{-}14)$	Ratio $\frac{q_i}{q}$	Average Air Temp. Room Center F t_a	Calculated Heat Loss Btu/hr $q_a = 112.5(t_a\text{-}14)$	Ratio $\frac{q_a}{q}$
2	32 in. C. I. Radiator	6672	67.3	5996	0.899	71.7	6491	0.973
16	Front Outlet Convector 13½ in. High	5723	67.2	5985	1.046	68.0	6075	1.062
19	Finned-tube Baseboard 8½ in. High	5946	65.3	5771	0.971	65.3	5771	0.971

was 55.2 F, and the rate of heat loss per degree of such temperature difference was 113.4 Btu per hour. This result is *greater* than the average for the twelve convectors, not *less*. His corresponding results for the baseboards are correct.

The calculations presented by Professor Fahnestock are used to support his suggestion that a low radiator under a window be used as a standard instead of a 32-in. radiator beside a window, as used in the paper. As an additional comment on that item, it may be pointed out that all radiators, convectors, and baseboards must be given consideration, even though some may have dimensions which will not allow a location under a window. Each heating unit must be installed in a practical manner and all must produce the required temperature at a selected index point. Effective heat factors are then introduced to give credit to a heating unit for producing the required temperature with a smaller heat input. The effective heat factors are necessary because the heat loss calculations cannot be refined to the point where they will rule out any of the poorer heating units. Since the poorest heating unit should just meet the heat loss requirements, it is logical to use it as a starting point for measuring heating effect factors as was done in the paper.

These tests, as well as many others reported to this Society, show that the rate of heat loss from a given structure depends upon the type of heating or heating unit installed. Other than ignoring this fact, possible procedures follow:

(1) Calculate the true heat loss and forget effective heat. This means calculating a different heat loss from the structure for each different type of heating system. The fact is that the indoor surface coefficient of heat transfer by combined radiation and convection is *not* 1.65 Btu per (hr) (sq ft) (F deg) between surface temperature and average air temperature for all heating systems in a given structure. This is equivalent to one of the procedures suggested by Mr. Trane.

(2) Calculate heat losses in the conventional, incorrect manner, as if they were independent of the type of heating system used and correct the observed test capacities of heating units through the principle of effective heat. In this paper, the numerical values of effective heat are based upon the arbitrary selection of a 32-in. cast-iron radiator as the standard heating unit. An alternate plan, that eliminates the need for selecting a standard unit, might be to compare the actual net rate of heat emission with a conventionally calculated heat loss. The choice of an index point arises in this procedure when an indoor air temperature must be selected for the heat loss calculation. Where in the room would the selected air temperature actually be found? Table D presents an expansion of this thought which was not presented in the paper.

1420

No. 1420

TRANSPORTATION HEATING AND AIR CONDITIONING SYSTEMS AND CONTROLS

By NORMAN O. KIRKBY*, CHICAGO, ILL.

ACCORDING to the present trend the American railroads are soon to be powered by locomotives other than steam. Therefore, the production of steam on diesel and electric locomotives must be substituted for the steam locomotive boiler as a source of heat, or heat generating equipment must be used on individual railroad cars. In addition, the distribution and control of the heating and cooling of railroad cars represent some of the most perplexing problems in the heating, ventilating and air conditioning fields.

The conception of the first passenger diesel locomotive immediately presented a major problem in heating. Heretofore the steam locomotive had been an almost inexhaustible source of heat, limited only by pressure and the size of the steam distribution lines. While a very large amount of heat was present on the diesel locomotive and wasted to the atmosphere through the radiators and exhaust manifolds, no practical method of harnessing this heat was, or is known, within the limitations of size and weight restrictions. As all railroad cars were heated by steam, some source of steam was necessary.

A vertical fire-tube boiler was first used but size and weight became a premium and it was necessary to find a new method of generating steam. The result was an entirely new concept of steam generation resulting in the first successful straight through steam generator. Such a generator had definite advantages over a conventional boiler. Since no water level is maintained and no water drums are used, the size of the water containing portions may be reduced to a minimum, consistent with high heat exchange practice. The small volume of water gave other advantages of safety. The mass of water in the coils at or near vapor temperatures is so small that a complete rupture of the coil may occur without any possibility of a steam explosion. Since cubical size was extremely limited, combustion volume was likewise limited and heat release per unit of combustion volume had to be increased greatly over standard practice. It was felt that such intensity could be achieved by burning in a high pressure chamber. Heat release on these generators is in the neighborhood of

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* Executive Engineer, Vapor Heating Corp.

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1,000,000 Btu per cubic foot of combustion space, approximately twenty times normal practice. Fig. 1 shows a typical steam generator now in use on diesel locomotives of American railroads. This particular steam generator has a normal rating of 4500 lb of steam per hour and is capable of a maximum output of 4800 lb of steam per hour.

These generators are continuous coil type steam generators, down-fired with the same diesel fuel as is used on the power engines. Sizes range from 500 lb of steam per hour to 4800 lb of steam per hour at varying pressures from 75 to 300 psig. Steam quality is in the neighborhood of 99.8 percent. Feedwater is pumped by semi-high speed triplex pumps to a water pressure regulator which bypasses water when steam pressure is within the control range. The water flows from the water pressure regulator to a hydraulic servo mechanism that

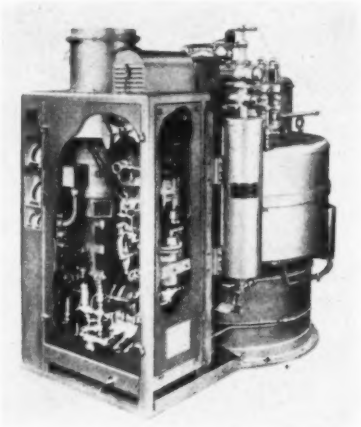

FIG. 1. 4500 LB PER HOUR STEAM GENERATOR

proportions fuel and air to the volume of water passing through the control. Here then is the major difference between a steam generator and conventional boiler. Water must be fed to the generator in exact accordance with steam demand, and practically all water entering the generating coils must be converted to steam in a single pass. No water level is maintained, therefore, flow controls must be used to regulate water, fuel and air. The volume of water flowing through the water portion of the servo control positions an oil valve which operates the hydraulic mechanism. Hydraulic pistons position a double cam plate to increase or decrease fuel and air. The double cam plate through a sleeve valve balances the hydraulic pistons to bring the system to equilibrium. The water then enters the steam generating coils where it is converted to steam. Any excess water is removed by a centrifugal steam separator to insure dry steam. The excess water, traveling at high velocities through the coils, carries the scale forming sludge into the separator where it is separated and wasted to the ground by automatic periodic blowdowns.

Atomization is by compressed air and a high pressure fan furnishes combustion air. A steam temperature limit control meters oil to the burner nozzle if lack of water causes superheating. Standard safety controls are used and *A.S.M.E.* practices and codes are observed throughout the manufacturing procedures. Thermal efficiencies are approximately 82 percent. Fig. 2 shows a typical schematic piping diagram of the type of steam generators now generally in use on American railroads.

The rules governing engineers and firemen prohibit the firemen from leaving the cab of the locomotive when the train is in motion. Therefore, all controls must be entirely automatic and in addition, remote alarms are used to indicate any failure of the generator or its appliances. Shut down of the unit upon arrival at terminals is accomplished by remotely operated valves which shut

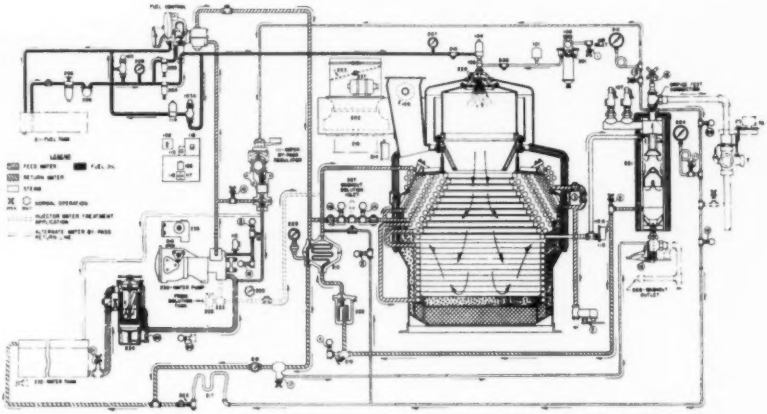

FIG. 2. SCHEMATIC PIPING DIAGRAM—STEAM GENERATOR

off the main steam line and, in turn, cause the generator to reduce its output to zero.

Periods of maintenance may well be irregular and at extended intervals, some railroads covering 5000 miles of operating before bringing the diesel locomotive into a terminal for maintenance. For this reason preventive maintenance is practiced and routine work highly systemized.

Other sources of heat have been used for railroad cars, some aimed at relieving the diesel locomotive from the problems of carrying water necessary for the production of steam and the additional fuel necessary for the steam generator. Others have been the outgrowth of electrical capacity problems. The advent of the modern high speed, lightweight, passenger car has brought increased competition between neighboring rail lines. More and more consideration is given to passenger comfort and increased use of electrical equipment has been the result. Capacities of air conditioning equipment have increased from 5 and 6 tons to from 7 and 8 tons per car. Better lighting and more of it has been used.

Radios, record players and intercommunication systems have imposed increased electrical loads. Diners are being built with entirely electrical kitchens. This constant increase in electrical load has increased the sizes of axle driven electrical generators to their limit. Axle load limits have been reached and alternate sources of electrical energy are being sought; on long transcontinental trains frequently a major portion of one diesel locomotive is being used just to drive axle generators and the locomotive contributes relatively nothing to the total tractive effort of the locomotive combinations.

In some cases head end electrical generating plants, diesel engine driven, have been used, but the problems of transmission between moving cars has not been satisfactorily solved.

More and more applications of individual engine generator sets on passenger cars are being made to solve the problems of increased electrical loads. Propane

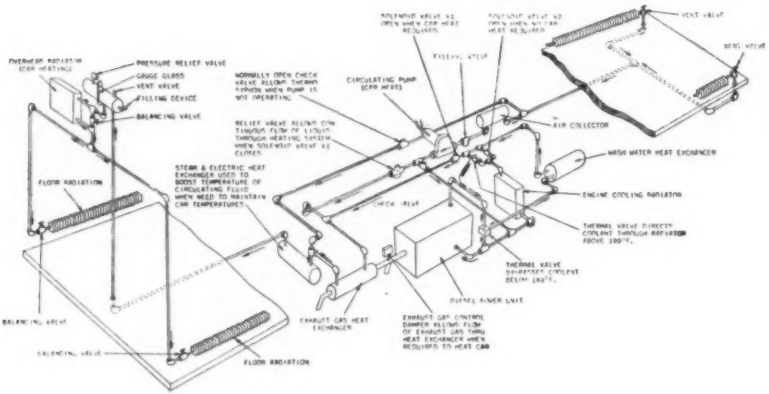

FIG. 3. SCHEMATIC DIAGRAM—WASTE HEAT RECOVERY

engines are used on some railroads for generating electrical energy and also for air conditioning. Much work is being done at the present time on diesel engine generator sets and in some of these cases, the waste heat from the diesel engine is being recovered. In some applications, the coolant water of the diesel engine is used as the heating medium within the car. In other applications, a liquid to liquid heat exchanger is used to heat water or a mixture of water and ethylene glycol which in turn is used as the heating medium. The largest single electric load is the air conditioning system; therefore, there is excessive capacity during non-air conditioning seasons. The energy may be used during the heating season as an auxiliary source of heat. Fig. 3 shows a typical schematic piping diagram of a waste heat system, using a diesel engine generator set placed underneath the car as the source of power and heat.

Thermostatic controls are arranged to use the available heat in predetermined sequence. First electrical current is used either by strip heaters in the overhead heating system or immersion heaters to heat the heating fluid. This insures maximum loading on the diesel engine. Next, full use is made of the heat wasted

to the coolant with thermostatic controls insuring constant engine manifold temperatures. Finally, the waste heat in the engine exhaust is used by means of a gas to liquid heat exchanger controlled by a bypass butterfly valve in the exhaust line. The sizes of the engine generator sets used range from 20 to 35 kw but full use of all recoverable heat will not satisfy the full heat load in colder climates and steam must still be used to supplement the engine heat unless an oil-fired water heater is used to make up any deficiencies. Such a heater as shown in Fig. 4 may be used alone as the heat source, or in conjunction with the heat from the axle generators or engine generator sets. Rated output is 150,000 Btu per hour at 8 gpm flow of liquid and up to 185,000 Btu per hour at flows of 20 gpm and upward, as normally used on passenger cars. Such heaters are also used to control automatically, through thermostats, the coolant

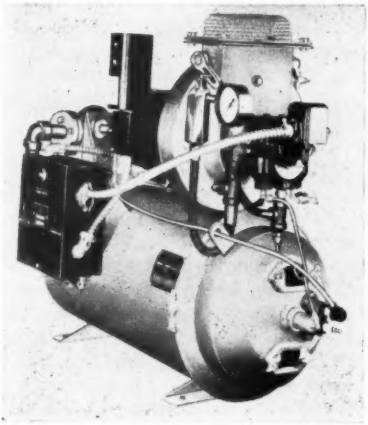

FIG. 4. 150,000 BTU PER HOUR WATER HEATER

water temperatures on diesel freight and switch locomotives during periods of non-use to eliminate the need for engine watchmen and to reduce maintenance costs resulting from prolonged operations at slow idling speeds.

Another solution to the problem of furnishing an individual source of heat for each car has been proposed and is being studied and tested at the present time. This is an offshoot of the so-called *vapor phase system* as applied to internal combustion engines. It consists generally of operating the engine at coolant manifold temperatures above the vapor temperatures at atmospheric pressures. The coolant flowing from the engine would be allowed to flash into steam and the steam condensed either in the heating system or undercar condensers. Higher engine efficiencies would result as well as increased overall thermal efficiency of the system. Pump heads seem, at the present, to be the limiting conditions when handling very hot liquids. There are also problems to be studied concerning the coring of the coolant water through the cylinder block of the diesel locomotives to avoid cavitation of the hot coolant liquids.

At present, however, approximately 95 percent of all passenger cars are heated with steam, and therefore its distribution and control are the major considerations of all railroad heating and air conditioning engineers. Fig. 5 shows a typical steam heating layout for an open section coach passenger car. High pressure steam is transported through the cars by 2½ in. main steam lines and through flexible metallic conduits between cars. Steam pressures vary from 125 to 250 psig. Total steam demand is greatly in excess of the total of the heat demand of the individual cars. The calculated heat loads of individual modern passenger cars at —20 F with a 40 mph wind is approximately 180,000 Btu per hour, or about 185 lb of steam per hour. Actual tests indicate steam

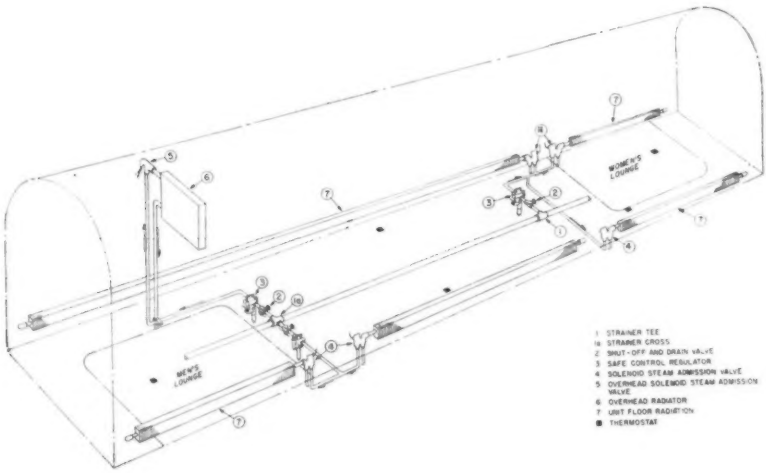

FIG. 5. HEATING LAYOUT—OPEN COACH

consumption may be as high as 350 to 400 lb of steam per hour per car. The difference in these figures is the loss due to main steam transmission lines and flexible metallic conduits, this loss increasing as temperature gradients increase.

The high pressure steam is reduced in pressure and fed thermostatically to the radiation at pressures from 3 to 12 lb. Condensate is wasted to the ground at temperatures of approximately 200 F.

Nowhere in industry does the heating and air conditioning engineer have more difficult problems to solve and more precise control specifications to meet. The space to be controlled is exposed on all six sides. It may be, and is propelled through space at speeds up to 100 mph. The same required equipment must be able to operate and control continuously from temperatures of 120 F in desert areas to —60 F in Canada or the Pacific Northwest.

Only one manual switch is used on the majority of passenger cars. This is a simple *Off* and *On* switch. Thermostatic controls determine whether the car will be heated, ventilated or air conditioned. A 1 deg ventilating dead band is usually used between heating and cooling, although cooling will extend into the

heating band. For example, a car will be heated to 75 F, ventilated to 76 F and air conditioned from 76 F, although the car may be cooled down to as low as 68 F. It is evident that controls must respond within less than ± 0.5 F to avoid hunting from heating to cooling and the reverse. An outside thermostat is used for positively locking out the compressor circuits when outside ambient temperatures are above a predetermined level.

CAR HEATING DESIGN

The heating system of a railroad car may be divided into three separate parts: an overhead forced air system, a side wall radiation system, and the radiant heating at the floor line.

The overhead forced air system is usually designed to take from 25 to 50 percent outside air, filter it, heat or cool it, and deliver it to the car through aspirating or diffusion type of outlets. With increases in air conditioning capacities, volumes of air have increased to avoid frosting of the evaporator coils. Air handled has increased from an average of 1500 cfm on prewar cars to 2400 cfm on postwar cars. Since duct sizes have remained relatively fixed, velocities have increased and air temperatures must be held fairly constant at the outlet to avoid hot and cold drafts. On prewar cars air intake hoods and filters increased in static pressure with the speed of the car and the result generally was 100 percent recirculated air at speeds of 50 mph and over. Since the war, intake hoods have been designed with a fairly even static head resulting in use of outdoor air at high speeds and increasing the total heat demand of the system.

If outdoor air intakes do not function at high speeds and exhaust fans are used, it is evident that a negative pressure might result in the car body and that is the case in a large proportion of the present railroad cars. Air leakage is increased tremendously and difficulty is encountered in keeping end doors closed. The problems imposed on the control system under these changes of air distribution require careful study and extreme accuracy to overcome. To overcome the problems of negative pressure and the resultant dirt problem, some cars have been pressurized to insure correct air distribution and cleanliness. The plenum chamber is sealed and no exhaust fans are used. Leakage stacks provide air outlets and air pressures of fractions of an inch are maintained throughout the car body.

All types of filters have been used including electrostatic, centrifugal and activated carbon. Those used most generally, however, are the wire mesh oil dipped type.

The increasing volumes of air handled in overhead systems have increased heating coil sizes from a prewar average of about 58,000 Btu per hour to about 100,000 Btu per hour on new cars.

Even with the increase in volume, velocity and heat input in the overhead system, tests prove conclusively that the overhead air stream cannot penetrate to the floor line of the cars without raising discomfort levels to unbearable limits. It must be remembered that conditions in railroad cars differ radically from those encountered in industrial or domestic service. Passengers sit next to, or in actual contact with, cold external walls and windows. Distances from air outlets to passenger contact are at a minimum. Outlets may be directly over the passenger at a distance of 5 ft from the head level when seated. The

inability to penetrate to the floor line necessitates the use of other heating systems to insure warmth at external walls and floor line.

A side wall heating system consisting of finned radiation is used to insure a vertical film of hot air along the side walls of the car. Such a system has been used for many years in railroad cars and while the full value of the system was not realized, its use became almost universal.

Before air conditioning became widely used, windows on all railroad cars were single glazed and also storm windows were used in cold weather. At that time, windows were approximately 40 percent of the side wall of the car adjacent to the passengers. These windows had large leakage factors and a high velocity vertical air film was the only method of counteracting such heat losses. Today windows make up approximately 80 percent of the car side wall exposed to passengers, but are double glazed and are well sealed in rubber. Radiant losses and gains have replaced the convectional losses and different methods are used to counteract these losses. In addition, heat distribution is better understood and may be more closely calculated.

Today the action of the side wall radiation is divided into two functions, that of convectional currents rising vertically along the outside wall and the radiant heat radiating in straight lines from the radiator guards in the form of infra-red rays.

It can be visualized then, that the air currents in the confined 10 ft wide space of a railroad car are at right angles to each other. A main air stream from the overhead system follows a path longitudinally of the car body and the convectional air currents from the side wall radiation move transversely and at right angles to the main overhead stream. These air movements become more complicated in multiple room cars but the main streams still retain their individual characteristics.

At one time it was felt that the convection currents from the side wall radiation kept the ankles and feet of passengers warm. It is now realized that the air currents moving from the aisle toward the side wall radiation must of necessity, be considerably cooler than the warm air delivered from the overhead system or the air rising from the side wall radiation. Today the value of infra-red radiation is fully recognized and indeed the raising of mean radiant temperatures to comfort levels has been a goal for some years. In 1934 the warm air from the side wall radiation was first used behind wainscot sheets being released at the window sill, thus giving a radiant shield between passengers and side walls and making effective a film of rising warm air on the inside of the windows. Today many cars are making use of such panel heated construction to improve heat distribution and increase the efficiency of the heating system.

In addition to the foregoing problems which must be solved by the control engineer, it has been found necessary to zone individual cars. End doors of the cars are constantly being opened, thus allowing cold air currents to enter the car body. This is especially objectionable at the non-recirculated end of the car where these cold air currents may easily affect the passengers adversely in the first two or three seats. For this reason, radiation at the end of the car is controlled separately by thermostats in that zone. In addition, lounge rooms for men and women are zoned separately as well as individual rooms in multiple room cars.

Additional zoning is used on many railroad cars which travel from east to

west across the country and from north to south on long trains. Here recognition is given to the differences between sunny and shady sides of the car, and consequently, thermostats on each side of the car regulate the amount of heat input to the car in accordance with radiant heat being added to that side of the car by sun effect. A particularly difficult problem in radiant effect is found in the so-called *dome cars,* now being used on many of the railroads. These cars are double-deck cars with the upper deck being in the form of a bulge at the roof line, the outer skin of the bulge being approximately 95 percent glass. Here radiant effect of sun can influence greatly the heating and air conditioning loads in the car, and thermostats sensitive to radiant heat are used to adjust the normal car thermostats to balance the heat input to the radiant heat imposed on the car structure.

Thermostatic control theories are fairly well defined although changes in seating arrangements and partition locations constantly must be analyzed anew. In general, it is well accepted that the overhead heating system must be fully used at all times to conform to the standards of ventilation set up by the U. S. Public Health Service. Thus the overhead system is used to heat the car until outside conditions, as indicated by inside temperatures, indicate the need for additional heat requirements from the side wall heating system. To accomplish this, side wall heating is tied in thermostatically to the overhead system and cannot be used except when a call for heat is recognized by the overhead heating thermostats. Such a system insures balance of heat and allows temperature levels to be maintained within 1 deg bands from end to end of the car and from floor to ceiling.

Bimetallic thermostats do not have the precision of control necessary for railroad requirements, nor will they stand the high vibration and shock conditions encountered. A mercury thermostat specially designed to meet these conditions is used almost universally for control. Where extremely cold temperatures are encountered, a mercury thallium amalgam is used in place of pure mercury in the thermostat. A unique feature of the thermostat is the fact that a secondary mercury bulb may be used. This bulb is wrapped with a resistance winding and insulated to be unaffected by ambient temperatures. The result is a thermostat which may be influenced by ambient temperatures or by artificial heat applied electrically through the resistance winding of the secondary bulb. The secondary bulb may be used in two ways. A steady amount of current may be imposed on the heat winding to set the thermostat down below its fixed temperature setting. Thus an 80 deg thermostat may be used as a 78, 76, 75 deg thermostat eliminating the need for an infinite number of fixed setting thermostats. By using a potentiometer in series with the heat winding, a controllable thermostat results. The heat winding on the secondary bulb may be used in another important fashion. By placing a resistance in parallel with the fixed resistance used for set down and having this resistance cycled by the relay operated by the thermostat, additional heat may be imposed on the secondary bulb when the mercury column is not in contact with the control point. When the mercury is in contact with the control point, a relay cuts out the cycling resistance and allows ambient temperature to affect the thermostat alone. By weighting the cycling resistance to the latent heat in the radiation being controlled, the heat input may be shut off at a point where the remaining latent heat in the radiation will bring the temperatures to the control point without overruns or underruns. While many

thermostats now use this feature, basic patents restricted its use to such mercury thermostats until recently.

Mercury thermostat-relay combinations control solenoid steam admission valves, electric strip heaters, oil fired heaters and, through interlocks, control the air conditioning equipment.

AIR CONDITIONING SYSTEMS

While electro-mechanical air conditioning equipment is most widely used, there are many cars equipped with steam jet systems and many which still use ice activated systems. It has been found that, generally, dry bulb control will give satisfactory results although reheat has been used in several ways. Overhead steam radiators have been used as well as electric strip heaters. In some

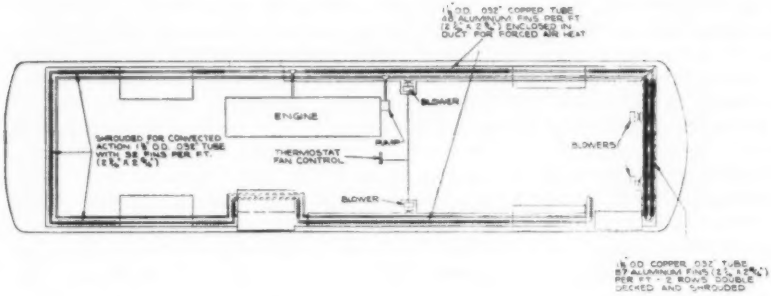

FIG. 6. BUS LAYOUT

Estimated Btu
Rear Section: no fans—16 ft at 850 Btu per foot—13,600 Btu
Side Sections:—25 ft at 2000 Btu per foot—Blowers on each side—50,000 Btu.
Front Section: 2 rows 57 fins per foot with 2 blowers—14 ft of radiation at 3500 Btu per foot
Total front end output: 4900 Btu
Total heat output: at 170 deg water, 112,600 Btu

cases reheat is under control of the occupant of a single room by means of a small electric heater in the duct outlet of the individual room. In addition, wet bulb control has been used in some cars and combination wet and dry bulb control has been used.

The same progress and forward thinking is being applied in other fields of transportation. In the bus industry more and more thought is being given to passenger comfort and modern heating and air conditioning controls are being applied. Normally the buses make use of the heat wasted by the engine to the cooling liquid for heating the interior of the bus. The coolant is pumped through various types of radiation, some in the form of unit heaters with fans to force air through the radiation core. In others, the radiation is distributed at the floor line and circulation is by convectional currents. Proper distribution of the heat uniformly throughout the bus body is being given more and more consideration in order to insure comfort levels at all seating positions in the bus. Fig. 6 shows a modern heating arrangement for a typical bus. While under normal operating conditions, the heat rejection to the coolant liquid by the bus engine is sufficient

to heat the bus properly, there are periods during which heat rejection is at too low a level to insure comfort within the bus body. This occurs at stopping points for loading and discharging of passengers and on long down hill runs where the bus engine is not being run. In some western states, these down hill runs may extend for lengths of 10 to 14 miles, with the bus engine operating not above idling speed, thus limiting the heat rejection to the coolant liquid and, in turn, the heat input into the bus. Some consideration has been given to the use of booster heaters for such periods when the heat rejection from the bus engine is not sufficient to make up the heat demand to the bus body. Oil fired or gasoline fired, small compact water heaters similar to that shown previously in Fig. 4 are used to make up the additional heat requirements during engine idling periods. These booster heaters also are used for standby protection for the bus during periods of non-use and when buses must be stored in outside storage lots during freezing weather.

Modern street cars of the PCC type make use of otherwise wasted heat from dynamic brakes and by means of thermostatically controlled dampers, either reject the heat or admit it into the car body.

AIRCRAFT APPLICATIONS

Aircraft equipment and controls, although necessarily more complicated and compact, also follow modern theories of control. Aircraft are heated and cooled by means of mixing dampers used to correct the proportion, cold and hot air, to satisfy temperature requirements in the cabin or body of the airplane. The source of the hot and cold air varies with various types of planes. In general, reciprocating engine driven airplanes use ram air as a cooling medium and ram air which has been heated by a gasoline heater for the hot source. The cold and hot air then is mixed to give a predetermined final temperature in the aircraft body. In general, cooling equipment is not used on commercial types of airplanes. The ram air cold source is sufficient to satisfy the cooling demands of the cabin or passenger carrying space. In military aircraft however and more especially in modern jet planes, it is necessary to mechanically cool the air so that a positive source of refrigeration is available at all times. On jet powered military aircraft, air is bled off from turbines and used as the hot source. A portion of this air is then fed through a cooling turbine where the air is forced to drive the turbine, giving up its energy and leaving the turbine as cold air. Here again the hot and cold air is mixed to satisfy the demand of the cabin. Thermostatic controls are now standard equipment on practically all aircraft for cabin heating and cooling, for de-icing purposes both on wings and cabin windows, for carburetor intake air heating, cowl flap controls and oil cooler controls on reciprocating engine type aircraft, and for cabin heating, de-icing and cooling on jet powered aircraft. Rate of change of temperatures in aircraft may be as high as 50 deg per second and here mass must be reduced to a minimum in the temperature sensing element. Electronic temperature controls are being used in increasing numbers, although some jet aircraft have cabin heating controls, using the double mercury bulb thermostat because of its low cost and simplicity.

CONCLUSIONS

It is evident that even though it is not a high mass production, highly competitive industry, the field of transportation heating, ventilating and air con-

ditioning has kept pace with the most forward thinking and has offered valuable contributions to the general knowledge in this field of engineering. The unexcelled policies of service to the country and public of the transportation industries will insure that this, as well as other fields of engineering endeavor, will continue to progress.

1421

No. 1421

WATER-VAPOR PERMEABILITY OF BUILDING PAPERS AND OTHER SHEET MATERIALS

By E. R. Bell*, M. G. Seidl*, and N. T. Krueger**, Madison, Wis.

SEVERAL methods are currently in use for measuring the water-vapor permeability of building materials. The work described herein was carried on during the summer of 1947 in cooperation with the Division of Standardized Building Codes and Materials, *Housing and Home Finance Agency,* to evaluate the water-vapor permeability of many building papers and other sheet materials according to *Tentative Standard Procedure* C214-47T, *American Society for Testing Materials,* and minutes of *A.S.T.M.* Committee C16. The procedure and minutes included a dry and a wet method, both of which were used in the present work. The materials tested were largely paper-base compositions treated in various ways to make them resistant to the passage of water vapor.

Materials for the tests were furnished by the manufacturers. Sufficient quantities of most materials were received to prepare several wet and dry pans. A large percentage of the papers and sheet materials were treated with asphalt, some with tar, a few with wax and asphalt, and some with an aluminum-pigmented coating or aluminum foil.

Test Apparatus

Chambers: Two chambers ordinarily used for drying or conditioning wood were prepared for the permeability tests.

A highly sensitive thermostat located midway between the fans controlled the heat input through the resistance heating coils. One set of heaters was placed opposite each of the two fans. Nearly constant cooling was provided by circulating city water through the cooling pipes. A wood-element hygrostat suspended from the ceiling maintained a nearly constant relative humidity by controlling the quantity of steam permitted to enter the chamber through small holes at

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intervals along the length of the humidifier pipe. Air was circulated by two 24-in. fans, each rotating at 1140 rpm and having a capacity of 5500 cfm free delivery.

The air circulation is indicated by the curved arrows of Fig. 1. The fans forced the air into the space in front of the racks and through the racks over the permeability test pans to the rear of the chamber. In the space behind the

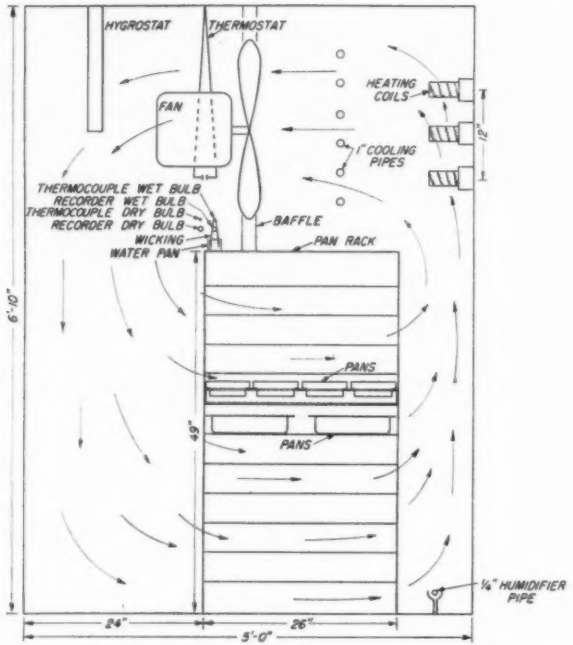

FIG. 1. SCHEMATIC CROSS SECTION OF VAPOR-PERMEABILITY TEST CHAMBER

racks the air was conditioned with respect to temperature and relative humidity before continuing through the fans.

Fig. 2 is a rear view of the chamber, showing some of the control mechanism and the recorder on which dry and wet bulb temperatures were recorded continuously. During the on-off cycles of heating and humidifying, the dry and wet bulb temperatures passed through a range of approximately 0.3 deg. The atmospheric conditions were maintained at 90 F and 50 percent relative humidity.

Racks: Each chamber was equipped with racks (Fig. 1) to support the pans and permit a free movement of air over the test specimens. The racks were 26 in. deep, so that two pans used in the wet method could be placed in each opening, one on the side of entering air and the other on the side of leaving

air. Six openings in each rack (Fig. 3) contained inserts, each of which held 10 dry pans. The racks were designed to provide an opening of ¾ in. above both the large and small pans. With this configuration, 120 dry pans and 108 wet pans could be placed in each chamber.

Pans: The aluminum pans used in the dry method and the copper pans used in the wet method are shown in Fig. 4. The aluminum pans employed in the

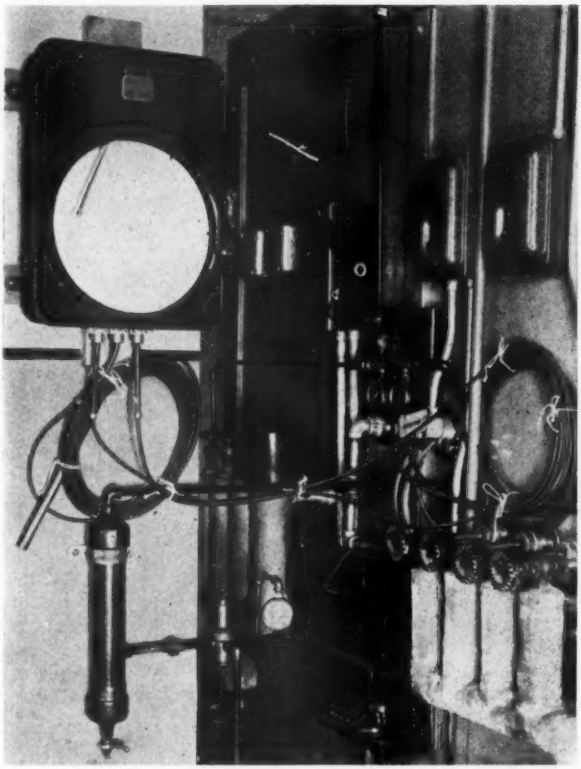

FIG. 2. REAR VIEW OF VAPOR-PERMEABILITY TEST CHAMBER SHOWING PART OF THE CONTROL MECHANISM AND THE WET AND DRY BULB TEMPERATURE RECORDER

dry method were 15/32 in. deep and had an exposed area of 20.6 sq in. The copper pans were 2 in. deep and had a ¾-in. horizontal flange in which to seal the specimens. The exposed area of the specimens was 100 sq in.

Anemometer: Velocities were measured with a temperature-compensated hot-wire anemometer. Calibration of the meter was checked by comparison with Pitot tube measurements in a wind tunnel at the College of Engineering, Uni-

versity of Wisconsin. Air velocities ranged from a minimum of 455 to a maximum of 1120 fpm and averaged 762 fpm over the sample surfaces.

Weighing Enclosure: Since it was considered highly important to weigh the pans in the atmosphere of the chamber, an enclosure was built inside the chamber to protect the pans from air movement. Fig. 5 shows the enclosure in place with one door open and a pan resting on the support ready to be weighed. A

FIG. 3. PAN RACK WITH ROUND DRY PANS IN WOODEN INSERTS AND SQUARE WET PANS PLACED DIRECTLY IN RACK

probe covered with sponge rubber, which extended into the enclosure from below, was used to arrest the motion of a pan within the enclosure before weighing. The enclosure was placed directly below an opening in the top of the chamber that permitted a fine wire suspension to be lowered from the arm of a balance on top of the chamber to the pan support in the enclosure.

Balances: Balances for weighing the dry and wet pans were placed on an adjustable platform on top of the chamber (Fig. 6). An analytical balance sensitive to 0.0001 g was used to weigh the dry pans. A triple-beam balance sensitive to 0.1 g was used to weigh the wet pans.

The microphone and amplifier visible in the background of Fig. 6 were part of an intercommunication system assembled for the convenience of the operators,

one of whom was inside the chamber to place the pans on the support in the weighing enclosure, while the other manipulated the balance.

Sealers: A mixture of 60 percent microcrystalline and 40 percent refined paraffin wax was used to seal specimens in the dry pans. The sealing medium for the wet pans was a rubber-base liquid adhesive that was used in conjunction with aluminum foil to define the test area and prevent edge leakage. The aluminum foil, which was mounted on wax paper for support, was cut into strips, each 1.5 in. wide and 14 in. long, one of which was used for each side of each pan.

Desiccant and Wet-Pan Filler: The desiccant used in the dry pans was anhydrous calcium chloride.

Wet pans were partially filled with water to provide approximately 100 percent relative humidity, sphagnum moss to provide a large surface for evaporation and to prevent surging of water when handling pans, and a fungicide to prevent

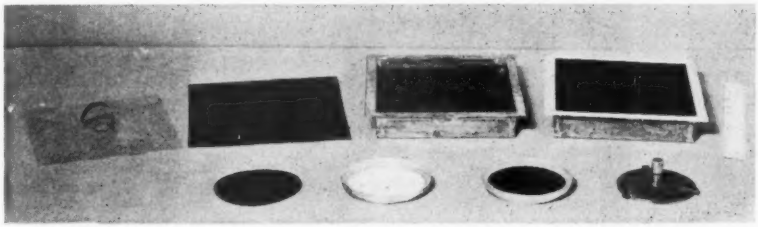

FIG. 4. COMPONENTS, TEMPLATES, AND COMPLETED ASSEMBLIES OF DRY AND WET PANS

(L. to R.) Top row: template, specimen, copper pan containing water and sphagnum moss, completed pan with specimen on top, and aluminum-foil strips. Bottom row: specimen, aluminum pan with calcium chloride, completed pan with specimen on top, template.

mold or fungus growth or chemical action that might evolve a gas and produce a pressure within the pan.

Templates: Templates to define the specimen area, to register this area properly with respect to the pan, and to aid in applying the sealing material are shown in Fig. 4.

TEST PROCEDURE

Because the materials to be tested were submitted in various quantities and forms, no standard sampling procedure could be followed. Some material was received in the form of 16- × 16-in. pieces, other material in rolls. The test specimens for any one material were obtained from widely scattered portions of the roll or from different 16- × 16-in. squares. In some cases, however, because of the insufficient amount of material supplied, no choice in the manner of cutting out the test specimens was permitted.

A dry pan was filled with calcium chloride granules to within 0.1 in. of the surface that supported the test specimen. The specimen, cut to the same size as the larger face of the template, was placed in the pan. A thin coating of zinc stearate powder was applied to the sloping edge of the template with a

cloth, in order to prevent the wax from sticking to the template. The template was placed on the specimen, and molten wax was poured into the annular space between the pan and the template. When the wax had cooled and solidified, the template was carefully removed and the voids left by the template guide arms

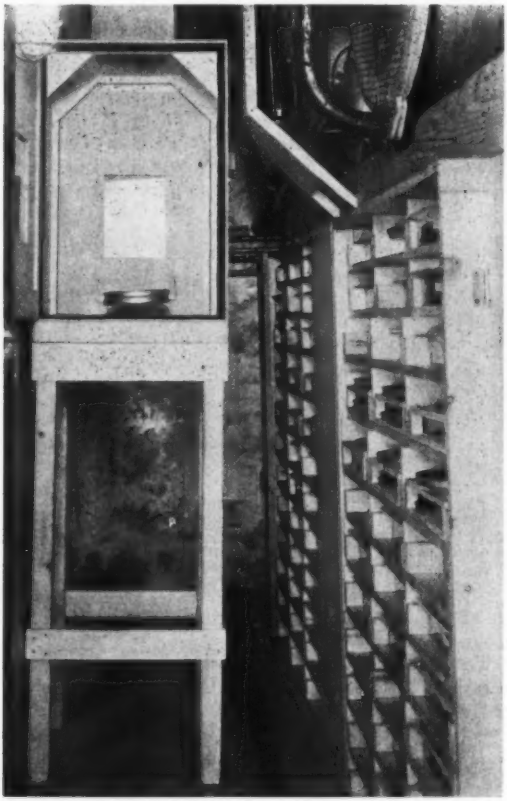

FIG. 5. INTERIOR OF VAPOR-PERMEABILITY TEST CHAMBER WITH WEIGHING ENCLOSURE AT LEFT AND RACKS AT RIGHT

One door of the enclosure is open to show a dry pan on its support, suspended by a fine wire attached to the analytical balance above the roof of the chamber.

were filled with wax. The pan was weighed and placed in the rack in the test chamber. Pans were weighed periodically until a constant rate of change of weight was observed. The permeability of the material was calculated from the constant rate of weight change.

A coat of adhesive was brushed onto both sides of the flange of a pan and allowed to dry at least an hour. The pan was filled with distilled water to within 1 in. of the test specimen, and enough sphagnum moss to cover the water surface was placed in the pan. A predetermined quantity of the fungicide solution was added to the water. A specimen was cut to the size of the pan, including the flange. The template was placed in the middle of the specimen, and the adhesive brushed onto the edge of the specimen extending beyond the template and allowed to dry for at least an hour. A second coat of adhesive was applied to

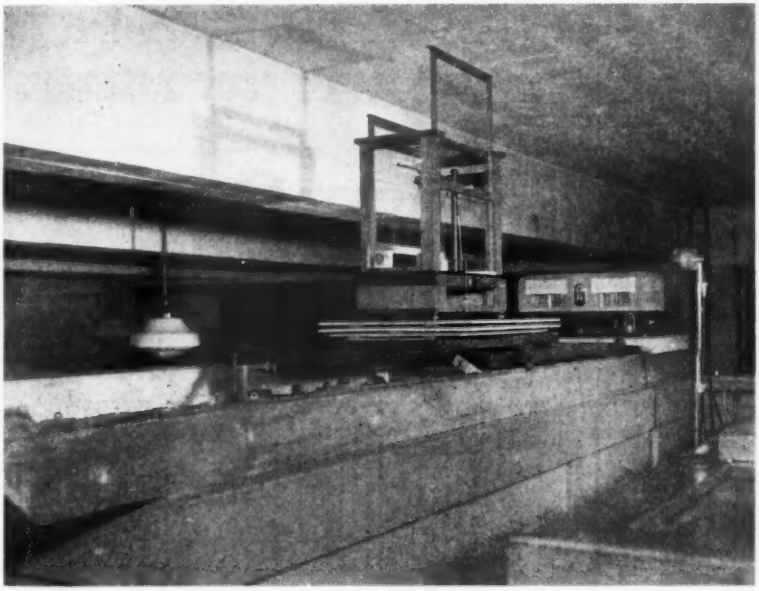

FIG. 6. ANALYTICAL BALANCE AND COMMUNICATION EQUIPMENT ON TOP OF THE CHAMBER

both flange and specimen, and the two were pressed firmly together 5 minutes after applying the second coat. The aluminum strips that covered the edges of the specimens were spread with adhesive, dried, respread, and applied with the same timing as was used for the test specimen. The pan was weighed and placed in a test-chamber rack. It was weighed periodically until its rate of change of weight became constant. Permeability was calculated from the constant rate of weight change.

The preparation of the dry pans deviates from *A.S.T.M. Tentative Standard Procedure* C214-47T only in the substitution of zinc stearate powder for petrolatum to prevent wax from adhering to the metal template. The zinc stearate has two distinct advantages: (1) it does not unite with any material likely to be tested by the dry method or the wax sealer, and (2) any of it that may

TABLE 1—WATER-VAPOR PERMEABILITY OF VARIOUS BUILDING MATERIALS DETERMINED BY THE DRY AND WET METHODS

Sample No.	Description of material[1] and remarks	Weight per roll of 500 square feet	Dry method: Number of specimens	Dry method: Water-vapor transmission[2]: Mean	Dry method: Water-vapor transmission[2]: Coefficient of variation[3]	Wet method: Number of specimens	Wet method: Water-vapor transmission[2]: Mean	Wet method: Water-vapor transmission[2]: Coefficient of variation[3]
		Pounds		Grains per sq. ft. per hr. per in. Hg.	Percent		Grains per sq. ft. per hr. per in. Hg.	Percent
	Single sheets sheathing and roofing products							
	Saturated with tar (tar felt)							
101	15-pound tar felt	70	3	1.560	35.2	3	9.355	2.9
1201	30.....do	139	6	33.304	16.9	5	36.323	4.6
1204	15.....do	70	3	4.055	1.7	5	18.234	8.5
	Saturated with asphalt (asphalt felt)							
102	15-pound asphalt felt	70	3	.971	23.6	3	5.564	2.4
103	30.....do	139	3	.498	7.8	3	3.465	8.2
1205	15.....do.....(Asbestos base)	70	6	.774	74.0	5	2.815	6.6
1206	15.....do	70	3	2.007	21.0	3	8.158	4.9
1207	30.....do	139	3	3.022	2.8	3	6.394	5.7
1212	Asphalt-saturated sheathing paper	22	6	3.267	21.2	3	20.231	7.5
1800	15-pound asphalt felt	70	3	1.207	2.7	3	5.661	8.5
1801	30.....do	139	6	.560	45.6	5	3.447	1.3
	Saturated and coated with asphalt (sheathing paper)							
400	Coated sheathing paper	43	5	.287	47.1	5	.566	19.9
1005	.....do	50	2	.146	8.9	3	1.046	13.8
1804	Coated sheathing paper, filler in coating	88	2	.156	26.8	3	.252	17.5
	Saturated and coated felts (roll roofing)							
100	Heavy saturated and coated felt	326	3	.052	34.5	3	.239	17.8
104	Heavy saturated and coated felt, one side glossy.....(Glossy side up)	200	2	.032	1.2	4	.143	9.7
	.....do.....(Glossy side down)		2	.033	10.6			
105	Heavy saturated and coated felt	260	3	.035	3.6	3	.158	13.3
106	Heavy saturated and coated felt, mineral filler in coating	200	3	.030	33.3	3	.130	14.8
1200	55-pound smooth roll roofing, mineral surface	255	2	.030	.0	3	.581	51.6
1202	45-pound smooth roll roofing, mineral filler in coating	210	3	.027	13.1	3	.366	47.1
1203	60-pound smooth roll roofing, asbestos base	270	3	.080	39.7	3	.293	44.9
1208	65-pound smooth roll roofing, mineral filler in coating	300	3	.027	21.6	3	.376	22.6
1802	65.....do	300	3	.017	.0	3	.122	69.1
1803	55.....do	255	3	.081	75.2	3	.180	71.2

Duplex or laminated products. Untreated kraft asphalt laminae not reinforced

602	20-20-20	11	2	.805	3.8	2	1.976	3.5
605	30-60-30	21	5	.527	16.3	5	1.660	33.4
609	30-30-30	16	8	.447	4.8	3	.768	4.8
1103	30-100-30	27	2	.280	.0	1	.451	
1123	60-30-60 (Insufficient material for wet pan test)	25	2	.850	1.6			
1116	60-60-60 do.	30	2	.300	1.3			
1211	60-80-60	35	3	.330	11.3	3	.743	5.3
1215	30-30-30	16	3	.846	13.4	3	2.917	1.8
1600	30-30-30	15	3	1.081	.7	3	2.014	11.9
1601	30-50-30	18				2	2.454	45.0
1602	60-60-60	30				2	2.398	37.7
1603	30-40-30	17	3	.711	6.1	5	1.752	30.5
1619	50-60-50	27	3	.526	7.7	2	.864	22.4
1700	(No description submitted)	43	3	.227	7.6	3	.876	15.6
1704	do.	15	3	.703	11.8	6	1.585	21.4
1706	do.	22	6	.323	14.7	3	.422	13.3
1709	do.	30	6	.504	34.2	3	1.349	29.7

Duplex or laminated products, untreated, creped

603	30-30-30	14	5	1.812	9.5	7	3.939	9.1
610	20-80-20	21	8	2.145	22.7	5	3.845	16.0
1100	30-60-30	20	2	.345	11.5	1	.925	
1101	30-30-30	15	2	.583	.2	1	1.547	
1104	30-30-30	15	2	.657	7.0	1	1.702	
1106	40-73-40	26	2	.840	10.7	1	2.391	
1110	30-30-60 (Insufficient material for wet pan test)	20	2	.884	20.1			
1111	30-60-30 do.	20	2	.504	24.5			
1213	40-55-40	23	6	3.159	53.9	6	6.184	39.9
1620	50-130-50	38	5	.205	26.4	3	.908	15.5

Duplex or laminated products, untreated, reinforced

1102	30-30-30	15	2	.305	5.2	1	2.194	
1105	30-85-30	24	2	.654	8.8	1	2.333	
1107	20-20-20	10	2	1.241	9.7	1	1.660	
1108	20-20-20	10				1	1.491	
1109	(No description submitted)					1	1.006	
1112	30-30-30 (Insufficient material for wet pan test)	15	2	.473	19.1			
1114	40-95-40 do.	30	2	.354	.3			
1115	30-80-30 do.	24	2	.296	15.8			

TABLE 1 (CONT.)—WATER-VAPOR PERMEABILITY OF VARIOUS BUILDING MATERIALS DETERMINED BY THE DRY AND WET METHODS

Sample No.	Description of material[1] and remarks	Weight per roll of 500 square feet	Dry method: Number of specimens	Dry method: Water-vapor transmission[2]: Mean	Dry method: Water-vapor transmission[2]: Coefficient of variation[3]	Wet method: Number of specimens	Wet method: Water-vapor transmission[2]: Mean	Wet method: Water-vapor transmission[2]: Coefficient of variation[3]
		Pounds		Grains per sq. ft. per hr. per in. Hg.	Percent		Grains per sq. ft. per hr. per in. Hg.	Percent
	Duplex or laminated products, untreated, reinforced (continued)							
1209	30-120-30	30	3	0.389	11.2	3	1.119	10.1
1210	45-90-45	30	3	.362	14.7	3	2.423	17.4
1501	30-120-30	34	3	.346	10.3	3	1.839	3.6
1606	30-90-30	25				2	1.612	21.5
1607	30-120-30	33				2	1.430	7.4
1608	60-150-60 Creped one side (Creped side up)	45	2	.274	4.0	3	1.257	5.4
	..do.. (Creped side down)		2	.399	9.2			
1609	60-180-60 Creped	50				2	1.197	3.3
1610	40-180-40 ..do..	43				2	.460	6.1
1612	30-120-30	30	3	.241	2.1	3	.835	6.5
	Treated duplex or laminated products. One or more sheets saturated with asphalt							
1214	60-30-60 Saturated one side (Saturated side up)	30	2	1.037	3.4	4	6.369	38.7
	..do.. (Saturated side down)		2	.935	20.6	4	5.741	12.9
1502	Saturated two sides, reinforced	39	6	.258	33.1	6	1.067	28.0
1604	50-65-50 Saturated one side	28				2	1.864	20.8
1611	30-100-30 Saturated two sides, reinforced	27	3	.279	7.8	3	1.024	6.4
	Duplex sheets with reflective coating - asphalt laminae							
1300	Coated two sides	15	3	99.230	13.5	5	76.615	5.1
1301	Coated two sides, reinforced	35	3	.226	8.1	3	.625	33.1
1302	..do..	47	6	.349	24.6	5	1.232	6.0
1503	..do..	40	3	.304	11.5	3	1.377	12.2
1614	20-80-20 Aluminum foil two sides	28	3	.230	88.7	3	.041	72.8
1705	Aluminum foil one side (Aluminum side up)	43	2	.002	141.5	2	.176	88.9
	..do.. (Aluminum side down)		1	.001	.0	2	.163	64.6
2000	..do.. (Aluminum side up)	49	2	.107	134.0	2	.331	24.5
	..do.. (Aluminum side down)		2	.012	126.5	2	.248	10.8
	Duplex sheets with miscellaneous treatments and coatings							
1500	Shrinkproof treatment both sides, reinforced	35	3	.294	6.4	6	2.031	14.3
1605	50-65-50 Abrasion-resistant treatment one side	28				2	4.873	19.0
1613	Aluminum foil - 20-pound asphalt-30-pound kraft (Aluminum side up)	18	2	.000	.0	2	.027	40.6
	..do.. (Aluminum side down)		2	.002	22.8	2	.228	41.1

	.....do.....(Aluminum side down).....		2	.002	22.8	2	.228	41.1
	Insulation back-up paper							
1000	Fire-resistant treatment one side (Treated side up).....	48	2	.253	17.3	8	1.774	37.5
	.....do.....(Treated side down).....		2	.116	18.0	8	3.612	33.3
1004	Paraffin-impregnated one side, asphalt side ribbed.....(Asphalt side up).....	18	3	.133	1.2	2	2.497	5.9
	.....do.....(Asphalt side down).....		4	.237	52.2	2	2.835	2.2
1616	Single-infused, ribbed.....(Asphalt side up).....	18	2	.232	25.7	2	.589	4.6
	.....do.....(Asphalt side down).....		2	.162	11.1	2	.446	8.6
1617	Single-infused.....(Asphalt side up).....	12	4	.443	44.6	2	1.801	8.1
	.....do.....(Asphalt side down).....		4	.583	81.3	2	.806	76.4
1806	Asphalt-saturated, one side glossy (Glossy side up).....	21	2	.327	35.6	2	4.234	94.2
	.....do.....(Glossy side down).....		2	.377	38.5	2	.583	5.2
	Single sheet - kraft - double infused							
606	Saturated kraft - 70 percent kraft - 30 percent asphalt...	16	5	30.764	19.9	5	41.926	7.1
607	Saturated kraft - 66-2/3 percent kraft - 33-1/3 percent asphalt.....	11	5	6.197	5.2	5	9.755	8.1
608	Saturated paper, creped - 80 percent kraft - 20 percent asphalt.....	9	3	99.323	5.2	5	67.807	4.6
1621	90-pound kraft saturated to 115-pound.....	19	3	99.345	1.3	3	66.997	2.2
1701	Asphalt-saturated kraft.....	18	6	3.612	63.5	6	9.429	35.2
1702	.....do.....	23	5	7.987	34.2	6	12.235	22.0
1703	.....do.....	11	6	7.447	17.3	6	11.916	9.6
1707	.....do.....	15	3	2.469	6.6	3	7.044	24.1
1708	.....do.....	22	6	.524	20.2	3	.976	6.9
1710	.....do.....	23	3	11.656	10.8	3	16.315	4.7
	Miscellaneous pulp and paper products							
1001	Single-sheet vapor barrier with fire-resistant coating two sides.....	48	9	.189	73.3	6	29.383	9.2
1805	0.00035-inch-thick aluminum foil on 20-pound bond paper.....(Aluminum side up).....	16	2	.005	93.1	2	4.370	1.4
	.....do.....(Aluminum side down).....		2	.003	141.4	2	1.072	11.7
2001	Corrugated asbestos on aluminum-foil-backed paper.....(Aluminum side up).....	49				2	.984	13.2
	.....do.....(Aluminum side down).....					2	1.684	10.2

[1] Descriptions furnished by manufacturers.

[2] Values obtained at 90° F. and 50 percent relative humidity - vapor pressure = 0.711 inch mercury - adjusted to 1 inch mercury by dividing by 0.711. Vapor pressure at 90° F. = 1.4219 inches of mercury (ASHVE Guide for 1947).

[3] The standard deviation divided by the mean times 100. This value is a measure of variability resulting from nonuniformities in material and variations in test conditions.

TABLE 2—WATER-VAPOR PERMEABILITY MEAN VALUES IN INCREASING NUMERICAL ORDER, EXPRESSED IN GR PER (SQ FT) (HR) (IN. HG)

Dry method				Wet method			
Permeability	Sample number	Permeability	Sample number	Permeability	Sample number	Permeability	Sample number
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0.000	1613 (U)[1]	0.362	1210	0.027	1613 (U)	1.660	605
.001	1705 (D)	.377	1806 (D)	.041	1614	1.660	1107
.002	1613 (D)	.389	1209	.122	1802	1.684	2001 (D)
.002	1705 (U)	.399	1608 (D)	.130	106	1.702	1104
.003	1805 (D)	.443	1617 (U)	.143	104 (U)	1.752	1603
.005	1805 (U)	.447	609	.158	105	1.774	1000 (U)
.012	2000 (D)	.473	1112	.163	1705 (D)	1.801	1617 (U)
.017	1802	.498	103	.176	1705 (U)	1.839	1501
.027	1202	.504	1111	.180	1803	1.864	1604
.027	1208	.504	1709	.228	1613 (D)	1.976	602
.030	106	.524	1708	.239	100	2.014	1600
.030	1200	.526	1619	.248	2000 (D)	2.031	1500
.032	104 (U)	.527	605	.252	1804	2.194	1102
.033	104 (D)	.560	1801	.293	1203	2.335	1105
.035	105	.583	1101	.331	2000 (U)	2.391	1106
.052	100	.583	1617 (D)	.366	1202	2.398	1602
.080	1203	.654	1105	.376	1208	2.423	1210
.081	1803	.657	1104	.422	1706	2.454	1601
.107	2000 (U)	.703	1704	.446	1616 (D)	2.497	1004 (U)
.116	1000 (D)	.711	1603	.451	1103	2.815	1205
.133	1004 (U)	.774	1205	.460	1610	2.835	1004 (D)
.146	1005	.805	602	.566	400	2.917	1215
.156	1804	.840	1106	.581	1200	3.447	1801
.162	1616 (D)	.846	1215	.583	1806 (D)	3.465	103
.189	1001	.850	1113	.589	1616 (U)	3.612	1000 (D)
.205	1620	.884	1110	.623	1301	3.845	610
.226	1301	.935	1214 (D)	.743	1211	3.939	603
.227	1700	.971	102	.768	609	4.234	1806 (U)
.230	1614	1.037	1214 (U)	.806	1617 (D)	4.370	1805 (U)
.232	1616 (U)	1.081	1600	.835	1612	4.873	1605
.237	1004 (D)	1.207	1800	.864	1619	5.564	102
.241	1612	1.241	1107	.876	1700	5.661	1800
.253	1000 (U)	1.560	101	.908	1620	5.741	1214 (D)
.258	1502	1.812	603	.925	1100	6.184	1213
.274	1608 (U)	2.007	1206	.976	1708	6.369	1214 (U)
.279	1611	2.145	610	.984	2001 (U)	6.394	1207
.280	1103	2.469	1707	1.006	1109	7.044	1707
.287	400	3.022	1207	1.024	1611	8.158	1206
.294	1500	3.159	1213	1.046	1005	9.355	101
.296	1115	3.267	1212	1.067	1502	9.429	1701
.300	1116	3.612	1701	1.072	1805 (D)	9.755	607
.304	1503	4.055	1204	1.119	1209	11.916	1703
.305	1102	6.197	607	1.197	1609	12.235	1702
.323	1706	7.447	1703	1.232	1302	16.515	1710
.327	1806 (U)	7.987	1702	1.257	1608 (U)	18.234	1204
.330	1211	11.656	1710	1.349	1709	20.231	1212
.345	1100	30.764	606	1.377	1503	29.383	1001
.346	1501	33.304	1201	1.430	1607	36.323	1201
.349	1302	99.230	1300	1.491	1108	41.926	606
.354	1114	99.323	608	1.547	1101	66.997	1621
		99.345	1621	1.585	1704	67.807	608
				1.612	1606	76.615	1300
(Continued in columns 3 and 4)				(Continued in columns 7 and 8)			

[1] U indicates "up" face, D "down" face, of numbered specimens, as described in table 1.

remain on the specimen can be removed with a light puff of air or a few strokes of a soft brush.

The method of preparing a wet pan, according to *A.S.T.M. Tentative Standard Procedure* C214-47T, is nearly the same as that for the dry pan. Pans used in the present work had a much larger area (100 sq in.) than the *A.S.T.M.* minimum requirement (about 10 sq in.). The sphagnum moss served to prevent surges of water when the pans were handled, and to provide a large surface for evaporation of water and thereby to insure a practically saturated atmosphere at the lower surface of the sample. The fungicide was added to prevent mold or fungus growth. The method of sealing previously described is quite different from the *A.S.T.M.* method. It is relatively simple, and tests proved it to be entirely satisfactory.

RESULTS OF TESTS

All of the materials tested are listed in Table 1 by groups. The table gives a description of each material, the mean vapor-permeability value, and the coefficient of variation for tests performed by both the dry and wet methods. Two values are given for those materials having faces that differ from each other. The face designated as *up* in the table was exposed to the atmosphere of the chamber. Although there is no satisfactory way to indicate the variability of a very small number of values, the coefficient of variation was chosen as the best means of expressing variations within the individual materials. It is expressed in percentage and determined as follows:

$(\sigma/\bar{x}) \times 100$ = coefficient of variation

$$\sigma = \sqrt{\frac{\Sigma x^2 - \frac{(\Sigma x)^2}{n}}{n-1}}$$

where

σ = standard deviation
x = individual value
$\bar{x}$ = mean
n = number of values

Table 2 gives the permeability values arranged in ascending numerical order. Values obtained by the dry method and by the wet method are presented separately.

There is a considerable range in the values of the coefficient of variation; where this value is large it is a good indication of non-uniformity of material. The variability found in the material tested would appear to warrant the use of a large number of samples. A large coefficient of variation would be relatively unimportant for material having a permeability value considerably below a specified maximum, but it would be highly important for a material having a permeability value near a specified maximum. In the latter case, a material would be questionable and perhaps rejected on the basis of its unreliability.

The various materials differed greatly in mean vapor-permeability values (Table 1), ranging from approximately 0 to about 100 grains per (sq ft) (hr) (in. Hg). The wet method yielded values nearly always greater than those determined by the dry method; moreover, the ratio of the two values was not

constant. The majority of the materials that had low permeability values by the dry method had low permeability values by the wet method. One of the several exceptions to this general trend was sample No. 1001 (single-sheet vapor barrier with fire-resistant coating on both sides), which had a low value, 0.189, by the dry method, and the relatively high value of 29.383 by the wet method.

It is evident that the wet method is a more rigid test than the dry method, because many materials that fell in classes A and B when tested by the dry method failed to meet specifications for these classes when tested by the wet method. All of the materials that fell in classes A and B when tested by the wet method fell in class A when tested by the dry method. Materials that did not fall in classes A, B, or D, were placed in class C, although no specifications were given for this class.

Further investigation is needed to determine the causes of the differences found between the two test methods, and whether materials acceptable by one method of test and not by the other offer adequate protection when used as a vapor barrier. A single standard procedure would be desirable for all sheet material, but it should be correlated with service tests to establish its validity as a permeability test.

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DISCUSSION

F. A. JOY, State College, Pa. (WRITTEN): I want to express my appreciation for the excellent job that has been done in the wide coverage of this field. The presentation of water vapor data by both dry and wet methods is especially commended since each figure has certain applications

At the same time, the practical use of this data by the architect or engineer appears to be quite difficult since the listed specification of a paper is not an accurate guide

to its performance. The objection to listing trade names is recognized, and in this dilemma it is urged that the manufacturer furnish with his product some statement of its vapor permeance expressed in uniform terminology readily understood.

In this connection, there is a growing realization that the use of the term *permeability*, though common, is unfortunate when applied to a sheet of paper. *Permeance*, like *conductance* and *resistance*, is a better term to use for any specimen of given thickness, such as paper. *Permeability* is a property of the substance and numerically equal to the permeance of a unit thickness. The unit of permeance used in this report—one grain per (sq ft) (hr) (in. Hg)—is desirable and widely used, but this rather complicated name seems to be quite unnecessary. The name *Perm*, as a convenient substitute, is finding increasing acceptance and perhaps will help to simplify and popularize the whole range of vapor transfer knowledge. We all speak familiarly of simple units like *ohms*, *Btu's*, etc., but may be somewhat disconcerted by the inclusion of mercury in a water vapor unit. With your indulgence, I will use these terms in what follows.

With few exceptions the tables show greatly increased permeance of a specimen when tested by the wet method. This is not surprising and is probably due in large measure to the naturally higher permeability of hygroscopic substances such as paper fibers at high relative humidity. It seems probable also in the wet tests that small temperature changes of the specimen will allow condensation upon its under surface, and the inflow of this liquid water into a capillary surface is of course rapid. Such condensation is expected when the specimen has low permeance insuring 100 percent relative humidity in the cup, and a temperature drop, though small, affects the specimen faster than the body of water below it. Deliquescence may also play an important part in water transfer when a specimen containing a water soluble material is exposed to high relative humidity. This action is probably illustrated most emphatically in Test No. 1001 where the paper has a *fire resistant coating*, and a wet method permeance 150 times the dry method value.

If, in its intended service, the paper is expected to be wet by condensation, or otherwise, it is proper to so test it. In fact, greater assurance of wetness should be provided. If, however, its permeance to vapor alone at a high relative humidity is wanted, it would seem desirable to eliminate any chance of condensation by lowering the relative humidity in the test cup a known amount. This can be done with an appropriate salt, using a saturated solution to insure a steady vapor pressure if desired. This technique we have used with good results.

The exceptions here reported to the general rule, *i.e.*, those papers having lower permeance by the wet method, deserve comment. There were only four. Three were in the highest range tested—100 perms—and include all tests in that range. This reversal at the 100 perm level may be more apparent than real, and its explanation may lie in the technique or calculation. The assumption of 100 percent relative humidity in wet cups, and 0 percent in dry cups is probably sufficiently correct when the specimen permeance is low, but in the high range it seems obvious that these atmospheres cannot be maintained close up to the under side of the paper. If the effective relative humidity in a wet cup test is considerably below 100 percent, such a reduction acts in three ways to diminish the results: (1) The average permeability of the paper fibers is less; (2) The chance for condensation is removed; (3) The calculation assumes a wrong vapor pressure difference.

This discussion of the three high permeance samples is important only as it tends to confirm a suspicion of the wet method as used, and as it demonstrates a need for consideration of the actual vapor pressure in the cup if the tested permeance value is to be considered other than nominal. The measurement of vapor pressure in cups is possible with an electric hygrometer.

The use of different cups for the two methods seems to need additional explanation, for it would appear simpler to use the same cup for both. Fundamentally, a smaller volume of water than of desiccant is required, and a shallow cup with means to

reduce water surges would seem adequate. An inorganic grid will do this and the desirability of moss may be questioned. It is doubtful if moss, or any other wick, increases evaporation in still air as long as the projected area is fixed. Moreover, it is doubtful if 100 percent relative humidity is a desirable goal.

These are some of the things that occur to me and may be of value in the interpretation of these manifestly important data.

R. H. HEILMAN, Pittsburgh, Pa. (WRITTEN): This is a very timely and important paper, and the authors are to be commended for the great amount of work which they have expended in obtaining the data for Tables 1 and 2.

I know that Committee C16 of the *A.S.T.M.* appreciate the extra effort expended in running all these tests by both the dry and the wet method. The wet method has been criticized and discontinued by some organizations as being unreliable. These tests confirm the observations made by other investigators, that the vapor transmission of numerous materials is many times greater when tested by the wet method as against the dry method for the same vapor pressure differentials. There are, for example, instances such as low temperature insulation, where the vapor barrier is sometimes operating below the dewpoint temperature. In these cases much more accurate estimates of the vapor transfer can be obtained from tests made by the wet method (even if they are considerably in error) than if the estimates were made from tests obtained by the dry method. It has also been found that the wet method sometimes shows up unfavorable characteristics that would not have been observed if tested by the dry method alone.

It is interesting to note the results obtained on the aluminum foil backed materials. Ordinarily one would expect practically zero water-vapor transmission if the foil was thick enough to have no pin holes. Sample 1613 shows zero transmission when tested by the dry method with the aluminum side up, while the same sample shows a high ratio when tested by the wet method with the aluminum side down. Somewhat the same type of results are obtained on sample 1805. However, sample 1614 shows much higher rates of transmission when tested by the dry method as against the wet method. Do the authors have any ideas as to what causes these inconsistencies? Was the aluminum foil too thin to give consistent results, or is there a possibility of leakage through the seal?

I fully agree that much additional work is necessary on this important subject.

S. KONZO, Urbana, Ill. (WRITTEN): The comprehensive experiments reported by the authors bring forth some interesting features. The observed deviations between the wet and dry methods prompt me to ask the question whether any observations were made of: (a) the presence of water droplets on the lower surface of the specimens, which might serve to establish a moisture migration through the material by capillary action, and (b) the temperature of the liquid in the wet method under steady state conditions.

It seems conceivable that the dry method is an evaluation of permeability due to vapor pressure alone, whereas the wet method might be giving a permeability value affected by both vapor pressure difference and liquid moisture travel. In this connection it might be of interest to devise an experiment in which the liquid is on top of the sample and a determination is made of the rate of flow of water that oozes through the membrane. Whether or not this is feasible I do not know, but it may shed some light on whether liquid moisture travel is significant in the permeability values obtained by the wet method.

I am somewhat bothered by the expression of the coefficient of variation as a percentage of the mean value. For example, sample 1614 having a mean value of 0.041 grains and a coefficient of variation of 72.8 percent has a standard deviation of only 0.030 grains. On the other hand, sample 1300 having a mean value of 76.615 grains and a coefficient of variation of 5.1 percent has a standard deviation of 3.91 grains. It is true that sample 1300 shows a smaller percentage variability than does

sample 1614, but the actual standard deviation for sample 1300 is roughly 100 times greater than the mean value for sample 1614. The question that arises is whether the method of expressing the variability is a fair one. I have no suggestions to offer in this connection, but I would appreciate a discussion of the method of presenting variabilities and particularly in connection with the magnitude of probable errors of observations.

The last question that comes to mind is in connection with the air velocities over the samples. It would be of interest to know the velocity gradients at the inlet and outlet faces of the test chamber from top to bottom. If a noticeable variation exists, would there be a possibility of locating a given sample at different elevations in the chamber and determining what effects, if any, are caused by differences in air velocities over the samples?

Certainly the avowed intention of the investigators to pursue the investigation further should be encouraged.

A. G. H. DIETZ*, Cambridge, Mass. (WRITTEN): The authors have carried on a much-needed piece of work in determining the permeability of these materials. There has been considerable uncertainty respecting the actual behavior of various building papers and other sheet materials as vapor barriers, with the consequence that users could not be certain of the performance of their installations. This paper helps not only to provide a good deal of information, but helps to indicate to what extent the *A.S.T.M.* tentative standard procedure applies and to what extent modifications seem desirable.

The last paragraph of the paper points up a factor of particular significance to the user of these materials. More information, and more correlation with service tests are needed to determine the limits of validity of the test as a permeability test the results of which can be projected to service conditions. It is to be hoped that such work will be undertaken. There is so much confusion and difference of opinion respecting the value of and the need for vapor barriers that anything which can be done to clarify this point will be highly welcome.

AUTHORS' CLOSURE (L. V. Teesdale)†: The greater permeability or permeance of materials tested by the wet method than by the dry method is due, no doubt, to the higher permeability of hygroscopic materials at high relative humidities as suggested by Professor Joy. We also found, as he suggests, that some condensation would collect on the under surface of the samples. In the case of asphalt papers the water was colored, indicating some evidence of deliquescence, but the characteristic straight-line relationship between permeability and time indicated that deliquescence was not a measurable factor in these tests.

Those samples having the highest permeance indicated higher permeance in the dry test than in the wet test, and this may be attributed primarily to the fact that the rapid transfer of vapor through the sample caused a lowering of the humidity on the opposite side of the sample. It might be mentioned here that, with the dry test method, very frequent weighings were required with samples of high permeance in order to obtain transmission values before the desiccant was affected by moisture.

The object in using the larger pans or dishes for the wet test was to avoid the hazard of wetting the sample with surging water. It is difficult to handle the small, shallow dishes when they contain water. The use of the sphagnum moss to increase evaporation and prevent surging of the water is simple and workable, though other methods of accomplishing the same objective may, of course, be used.

Where the test materials have the highest resistance—as, for example, those having aluminum foil surfaces—vapor transfer is probably limited to pin holes, and the num-

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ber and size of such microscopic openings may vary from sample to sample. This could explain the high coefficient of variation for samples having very high resistance values.

Close uniformity of air movement was difficult to attain, and a compromise was accepted where the minimum movement was 450 fpm or more and the maximum about 550 fpm. The maximum was exceeded by as much as 100 fpm in a few spaces. Extra samples were placed in these spaces and the transmission values compared with those of samples exposed to the lower air velocities. No measurable differences were found, indicating that where the average velocity is 500 fpm there may be considerable variation in air movement without affecting the transmission rate to a measurable degree.

The coefficient of variation (standard deviation/mean x 100) is a useful statistic for expressing variations since it is a ratio of two related averages having the same unit of measurement. While, in general, series of means and standard deviation tend to change together, unusual variances (s^2) usually are shown to be present by the coefficient of variations. In the case cited by Professor Konzo, the means of samples 1300 and 1614 are, respectively, 76.616 and 0.041 and their respective coefficients of variation are 5.1 and 72.8 percent. This shows that the relative standard deviation of 1300 is considerably smaller, although its mean and standard deviation are larger than the same statistics of 1614. Although these tests are limited in number, their coefficients of variation express differences in measurements of the sample that are not shown by either means or standard deviations. The differences of these samples appear to be due to difficulties in measuring the small permeability of sample 1614 that were not encountered in the much more permeable sample 1300. Therefore the coefficient of variation seems to be the most adequate way of expressing variation in these data.

1422

No. 1422

USE OF AIRCRAFT PROPELLERS FOR AXIAL FLOW FANS

By Elmer C. Lundquist*, Iowa City, Ia., and M. J. Hamilton†, Santa Monica, Calif.

AIRCRAFT propellers are frequently installed in ducts to be used as axial flow fans for moving large quantities of air against low pressures. The usual propeller performance curves are obtained from propeller tests in an unconfined air stream and are not suitable for predicting the performance of a propeller when acting as an axial flow fan. This paper will present results obtained from model tests on the performance of aircraft propellers used as axial flow fans with particular emphasis on the method of using these results to predict the performance of prototype fans of geometrically similar shape.

Data for two-, three-, and four-bladed propeller fans obtained from scale model tests will be presented in dimensionless form. Equations employing consistent units will be developed using the principle of dimensional analysis. Since equations in this form, however, are not commonly employed by the heating and ventilating engineer, the equations will be converted to a form incorporating conventional units. It will then be shown how the equations and data herein presented may be used to select a propeller fan of appropriate size and speed for any specified requirement of pressure rise and air capacity. It will also be shown how the performance of fans of any size geometrically similar to those tested may be predicted for any set of operating conditions.

Analysis of Problem

A preliminary dimensional analysis of the variables involved in the problem of determining the performance of aircraft propellers, when used as axial flow

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fans, provides useful information relative to the grouping of variables. It will be assumed that it is desired to determine the pressure rise produced by the propeller and the power required to turn the propeller as functions of the other variables involved. The variables that appear to be of importance in this problem are:

p = pressure rise across the propeller disc, pounds per square foot.
P = shaft power supplied to the propeller, foot pounds per second.
V = average axial velocity of flow through the propeller disc, feet per second.
n = rotating speed of propeller shaft, revolutions per second.
D = diameter of propeller, feet.
ρ = mass density of medium flowing through the fan, slugs per cubic foot.
μ = absolute viscosity of medium flowing through the fan, slugs per foot second.
a = velocity of sound in the medium flowing through the fan, feet per second.
β = blade angle of the propeller blade at a designated radius (in this case taken at a radius of 75 percent of the tip radius), degrees (dimensionless).

The shaft power required for the propeller fan can be expressed symbolically as

$$P = f_1 (V, n, D, \rho, \mu, a, \beta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

According to a theorem commonly employed in dimensional analysis a function such as the one expressed by Equation 1 can be reduced to a function involving a number of dimensionless terms equal to the number of variables involved minus the number of fundamental dimensions. All the quantities in Equation 1 may be expressed in terms of the three fundamental dimensions of mass, length, and time. Hence, the number of dimensionless terms necessary to express the relation between the variables will be $8 - 3 = 5$. Employing the procedures used in the theory of dimensional analysis, the following grouping of variables is obtained:

$$\frac{P}{\rho n^3 D^5} = f_2 \left[\frac{V}{nD}, \frac{\rho n D^2}{\mu}, \frac{nD}{a}, \beta \right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In a similar fashion, the pressure rise across the fan disc may be written as

$$p = f_3 (V, n, D, \rho, \mu, a, \beta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

which in turn can be reduced to

$$\frac{p}{\rho n^2 D^2} = f_4 \left[\frac{V}{nD}, \frac{\rho n D^2}{\mu}, \frac{nD}{a}, \beta \right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Both equations 2 and 4 involve five dimensionless terms, one of which is expressed as a function of the other four, and they imply that the value of the dependent variable is determined by fixing the values of the four independent variables. Furthermore, the value of the dependent variable is not affected by a regrouping of values of the terms making up the independent variables so long as the numerical values of the groups comprising the independent variables are unchanged.

As an example, suppose $V = 50$ fps, $D = 5$ ft and $n = 10$ rps. Then $V/nD = 1.0$. The value of $p/\rho n^2 D^2$ obtained experimentally for these conditions could be expected to be the same as for a fan geometrically similar in all respects

where $V = 30$ fps, $D = 3$ ft, $n = 10$ rps and $V/nD = 1.0$ provided, of course, that in both cases the values of $\rho nD^2/\mu$, nD/a, and β were identical.

Equations 2 and 4 both contain four independent variables. The experimental work which would be required to express either of these functions, by varying only one parameter at a time, would therefore be quite tedious. Fortunately it can be shown that two of these four variables contribute very little to the value of the function and can, in general, be neglected. These two variables are $\rho nD^2/\mu$ and nD/a.

The term $\rho nD^2/\mu$ is essentially the Reynolds number of flow around the propeller. The product of nD is proportional to the tip speed of the propeller, V_t and hence $\rho nD^2/\mu \cong \rho V_t D/\mu$. This latter term may be recognized as the familiar Reynolds number. Experience with the effect of Reynolds number for flow around airfoils indicates that the value of the coefficients of lift and drag are influenced only to a small extent by a change in Reynolds number. In general, the maximum value (the stalling value) of lift coefficient is increased as Reynolds number is increased but for low angles of attack the lift coefficient is independent of Reynolds number. Also, only the minimum value of drag coefficient is influenced by a change in Reynolds number. Since the blade elements of a propeller type fan are not operating at either the stalling or the minimum drag condition when the fan is operating near peak efficiency, and since the pressure rise and the shaft power required are both related to the lift and drag coefficients of the propeller elements, it can be concluded that the Reynolds number in this case represents a variable of small importance.

The term nD/a appearing in Equations 2 and 4 is the Mach number. Since $nD \cong V_t$, the tip velocity, the term nD/a is proportional to V_t/a which is the ratio of the tip speed of the propeller to the speed of sound. This is a variable that is unimportant in this analysis unless it attains a value in excess of from 0.7 to 0.8. For values of Mach number in excess of this, compressibility effects cannot be neglected. For lower values, however, the influence of a variation in Mach number has practically no bearing on the fan performance.

Eliminating the Reynolds number and the Mach number from Equations 2 and 4 results in

$$\frac{P}{\rho n^3 D^5} = f_5\left[\frac{V}{nD}, \beta\right] = C_Q \quad \ldots\ldots\ldots\ldots \quad (5)$$

$$\frac{p}{\rho n^2 D^2} = f_6\left[\frac{V}{nD}, \beta\right] = C_p \quad \ldots\ldots\ldots\ldots \quad (6)$$

where

C_Q = a power coefficient.
C_p = a pressure coefficient.

Both are dimensionless and are functions of the dimensionless quantities of V/nD and β.

The basic equations preparatory to experimental work have now been presented. The performance of the fan can be completely determined by obtaining the variation of C_Q and C_p with V/nD, known as the advance-diameter ratio, for various constant values of β, the propeller blade angle. Results obtained from model experiments are applicable to larger geometrically similar prototypes operating at the same value of V/nD and β. It should be remembered,

however, that this conclusion presumes dissimilarity in Reynolds number and Mach number to be of negligible importance.

EXPERIMENTAL APPARATUS

A schematic diagram of the test apparatus is shown in Fig. 1. A model airplane propeller 24¾ in. in diameter was mounted on a drive shaft in a duct 25 in. in diameter as shown. The propeller shaft was supported in bearings suspended by three radial rods or spokes anchored to the periphery of the fan casing. Tests were made on propellers having two, three, and four blades. In all cases, the propeller hub was not faired nor were stationary guide vanes installed.

A honeycomb straightener was installed in the duct six diameters downstream from the propeller. Pitot tube traverses, both horizontal and vertical, were

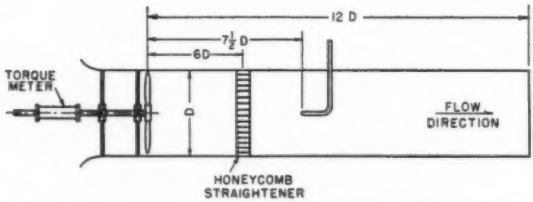

FIG. 1. SCHEMATIC DIAGRAM OF TEST EQUIPMENT

made at a point 7½ diameters downstream from the propeller. The average static pressure obtained at the traversing station was corrected to obtain the static pressure at the fan outlet by the methods recommended by the standard fan test code.[1]

The power input to the propeller fan was obtained by measuring the shaft speed and the shaft torque with the torque meter indicated in Fig. 1. This torque meter included a long, slender torque rod enclosed in a double barrelled casing. The outer cylindrical casing was mounted concentrically with the inner casing on ball bearings. The two casings were then restrained from rotating with respect to one another by the long torque rod. The application of torque would cause an angular deflection of the one casing with respect to the other, the torque being absorbed by the torque rod. When the fan was in operation the angular deflection was observed with a stroboscope. Since the torque meter was calibrated, this angular deflection could be translated into torque. The torque required to turn the propeller shaft without the propeller was measured at various speeds. This tare torque was subtracted for subsequent propeller tests to obtain the net torque required for rotation of the propeller.

The model propeller blades used in this investigation had the geometry of a typical aircraft propeller. The propeller shape is defined by the parameters given

[1] Standard Test Code for Centrifugal and Axial Fans (A.S.H.V.E. and the *National Association of Fan Manufacturers, N.A.F.M. Bulletin* No. 103).

in Fig. 2 in terms of the pitch-diameter ratio u/D, the blade width-diameter ratio b/D, and the maximum thickness-blade width ratio h/b. This nomenclature will be clarified by reference to Fig. 3.

The RAF-6 airfoil section was chosen as the basic section because its flat bottomed surface facilitated construction and pitch measurement, and because of its past application to propellers. As will be noted from Fig. 2, the propeller blades were designed so the pitch-diameter ratio u/D would be constant over

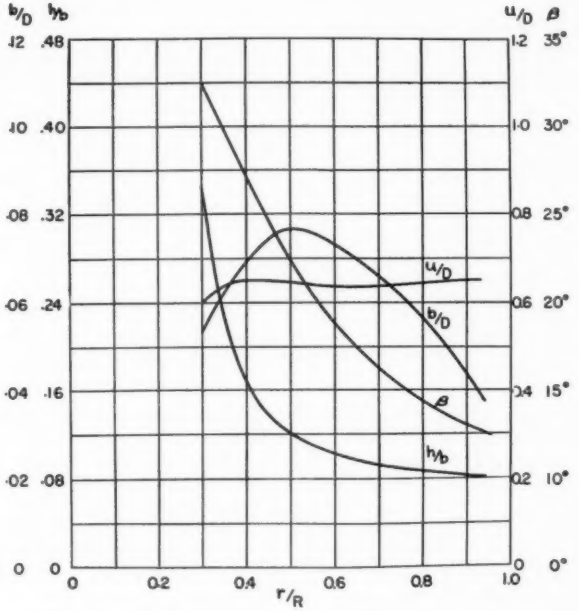

FIG. 2. PARAMETERS DEFINING PROPELLER BLADE GEOMETRY

most of the propeller blade when the blade angle was set at 15 deg at the radius $r/R = 0.75$.

The propeller blades used were mounted in the hub so the blade could be set at any blade angle desired. It is apparent that if the entire blade were rotated on its own axis, the blade angle would be changed by the same amount along the entire radius. Hence, for any blade setting other than the one for which Fig. 2 is drawn ($\beta = 15$ deg at $r/R = 0.75$) the distribution of u/D with respect to r/R would change. Subsequently, when blade angle is referred to, it is the propeller blade angle at the radius $r/R = 0.75$ and will be designated by β.

ANALYSIS OF RESULTS

Tests were run with two-, three-, and four-bladed propellers at blade angles varying from 7½ to 20 deg by 2½ deg increments. Enough measurements were taken for each test so that the air density, the fan rotating speed, the shaft power input to the fan, the static pressure at the fan outlet, the velocity pressure at the fan outlet and the air velocity at the fan outlet could be determined. For each propeller (two, three, or four blades) and for each blade angle, tests were run using several orifices of different diameters installed on the outlet end of the duct shown in Fig. 1. In this manner it was possible to obtain values of C_Q and C_p for a range of values of V/nD.

Performance charts giving the results of these tests are shown in Figs. 4 to 9 inclusive. In Figs. 4 to 6 inclusive the static pressure coefficient, C_{ps}, is plotted

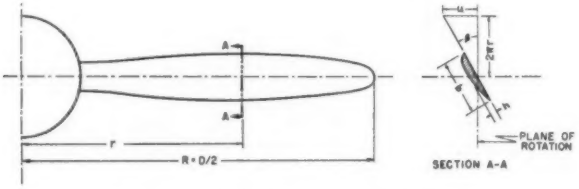

FIG. 3. PROPELLER BLADE DETAILS

as a function of V/nD and β. The static pressure coefficient is defined by

$$C_{ps} = \frac{p_s}{\rho n^2 D^2} \quad \ldots \ldots \ldots \ldots \ldots \quad (7)$$

where p_s is the static pressure increase in pounds per square foot through the fan and all other terms are as previously defined. The total pressure coefficient, C_{pt}, is plotted as a function of V/nD and β in Figs. 7 to 9 inclusive. This coefficient is defined as

$$C_{pt} = \frac{p_t}{\rho n^2 D^2} \quad \ldots \ldots \ldots \ldots \ldots \quad (8)$$

where p_t is the total pressure rise through the fan in pounds per square foot. The total pressure is defined as the sum of the static pressure increase through the fan and the average velocity pressure at the fan outlet.

Equations 7 and 8 have the same form as Equation 6. It should be evident from the development of Equation 6 that the equation is valid using either static pressure or total pressure.

The power coefficient C_Q is not shown in Figs. 4 to 9, but rather fan efficiency contours are plotted. Fan efficiency is defined as the ratio of the fluid power to the shaft power and may be expressed by

$$\eta = \frac{pAV}{P} \quad \ldots \ldots \ldots \ldots \ldots \quad (9)$$

where

A = the fan outlet area in square feet.

Substituting values of P and p from Equations 5 and 6 this equation becomes

$$\eta = \frac{C_p \rho n^2 D^2 (\pi D^2/4) V}{C_Q \rho n^2 D^5} = \frac{\pi}{4} \frac{C_p}{C_Q} \frac{V}{nD} \quad \ldots\ldots\ldots \quad (10)$$

Equation 10 may be used to define either static efficiency (η_s) or the total efficiency (η_t) depending on whether the pressure coefficient is C_{ps} or C_{pt}. Lines of constant efficiency, or efficiency contours, are shown in Figs. 4 to 9. These lines represent static efficiency η_s in Figs. 4 to 6 and total efficiency η_t in Figs. 7 to 9.

The shaft power in foot pounds per second may be computed from Equations 5 and 10. From Equation 10

$$C_Q = \frac{\pi}{4} \frac{C_p}{\eta} \frac{V}{nD} \quad \ldots\ldots\ldots\ldots \quad (11)$$

Substituting this in Equation 5 gives

$$P = \frac{\pi}{4} \frac{C_p}{\eta} \frac{V}{nD} (\rho n^3 D^5) \quad \ldots\ldots\ldots\ldots \quad (12)$$

It should be noted that $C_{ps}/\eta_s = C_{pt}/\eta_t$ for a fan having a given number of blades operating at specified values of blade angle and V/nD. Hence, Equation 12 may be employed using either the static or the total pressure coefficient so long as the corresponding efficiency is used.

It was previously pointed out that the test results obtained from model tests might be expected to apply to geometrically similar prototypes. Therefore, it may be expected that the performance charts shown in Figs. 4 to 9 may be applied in the selection of aircraft propellers of similar shapes for use as fans. In these figures, curves for blade angle increments of 1 deg are plotted. These were obtained by a system of cross-plotting the experimental data which were obtained for blade angle increments of 2½ deg.

USE OF PERFORMANCE CHARTS

In the application of a fan to a duct system the pressure the fan must produce is a function of the air velocity and the geometry of the duct system. For the system in which the air is not recirculated the fan must supply the energy to produce the velocity pressure as well as the pressure loss due to friction. The total pressure rise, p_t, required of the fan, then, must be given by

$$p_t = \frac{\rho V^2}{2} + f \frac{L}{D} \frac{\rho V^2}{2} = \left(1 + f \frac{L}{D}\right) \frac{\rho V^2}{2} \quad \ldots\ldots \quad (13)$$

The friction factor f is, in general, a function of the Reynolds number of the flow through the duct system. A commonly used value of f for flow through galvanized sheet metal duct work is 0.02 although this coefficient does vary somewhat with Reynolds number and with surface roughness of the duct. L is

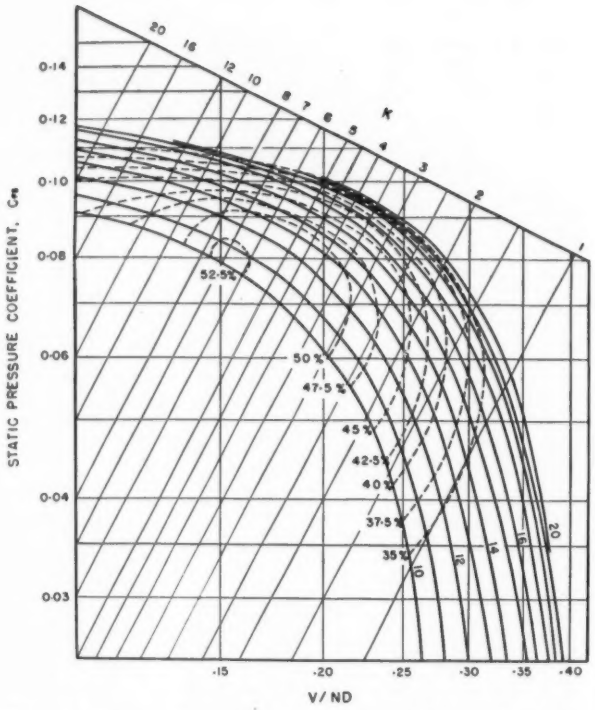

FIG. 4. STATIC PRESSURE COEFFICIENT C_{ps} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *two-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. STATIC EFFICIENCIES SHOWN IN CONTOUR

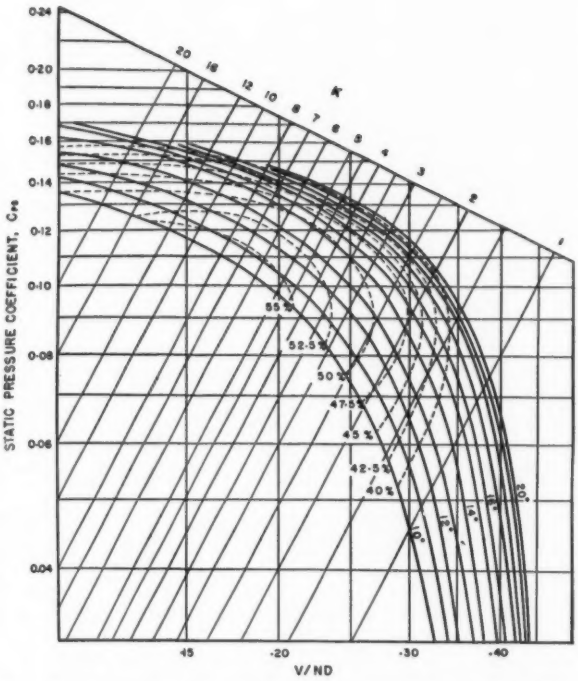

FIG. 5. STATIC PRESSURE COEFFICIENT C_{ps} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *three-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. STATIC EFFICIENCIES SHOWN IN CONTOUR

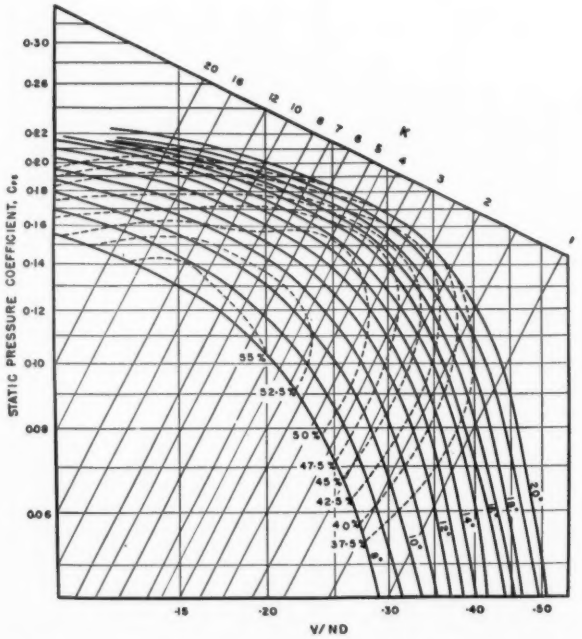

FIG. 6. STATIC PRESSURE COEFFICIENT C_{ps} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *four-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. STATIC EFFICIENCIES SHOWN IN CONTOUR

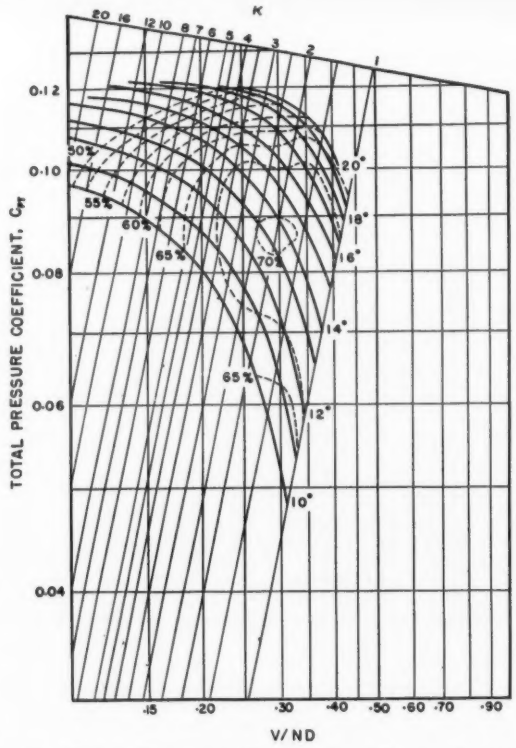

FIG. 7. TOTAL PRESSURE COEFFICIENT C_{pt} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *two-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. TOTAL EFFICIENCIES SHOWN IN CONTOUR

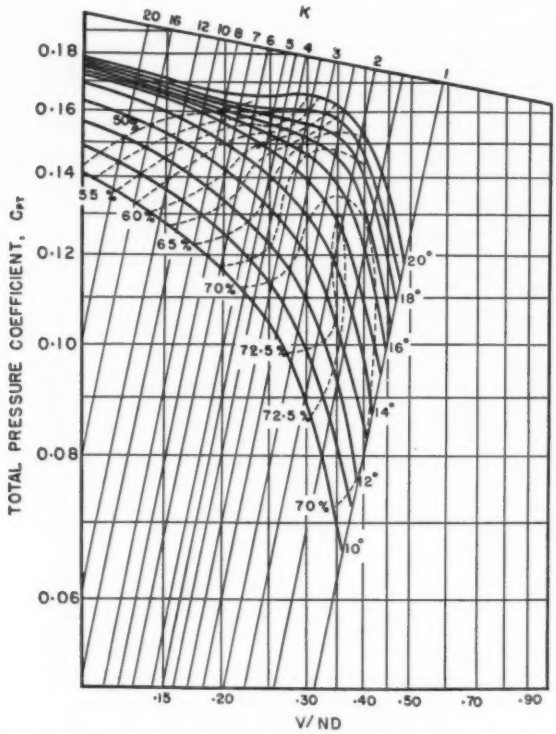

FIG. 8. TOTAL PRESSURE COEFFICIENT C_{pt} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *three-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. TOTAL EFFICIENCIES SHOWN IN CONTOUR

the length of straight duct of diameter D in which the same friction loss would be produced as in the actual duct system.

For the purpose of convenience, a pressure drop coefficient K is now introduced and will be defined for a non-recirculating system as

$$K = 1 + f\frac{L}{D} \quad \ldots\ldots\ldots\ldots \quad (14)$$

This coefficient K is fixed if the fan diameter D and the rate of airflow through the given duct system are specified.

Substituting the value of p_t from Equation 13 in Equation 8 gives

$$C_{pt} = \frac{K\rho\, V^2/2}{\rho n^2 D^2} = \frac{K}{2} J^2 \quad \ldots\ldots\ldots\ldots \quad (15)$$

where $J = V/nD$. Since Figs. 7 to 9 inclusive are log-log plots of C_{pt} *vs.* J it is apparent that Equation 15 would appear as a straight line on these charts

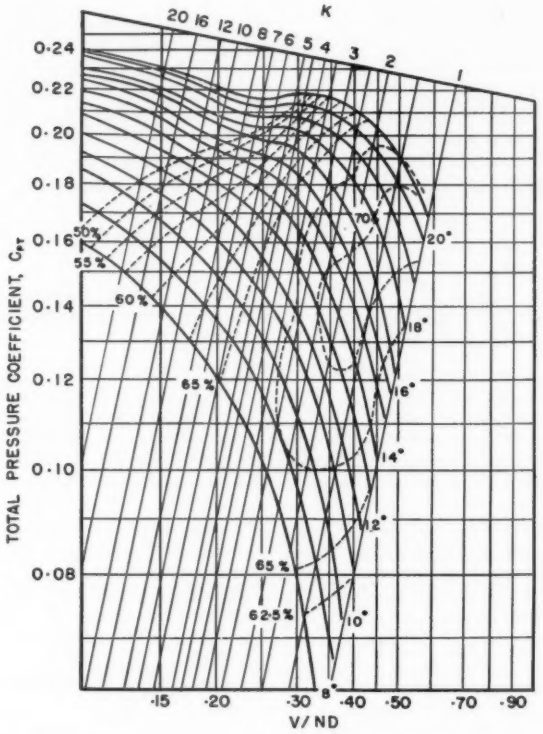

FIG. 9. TOTAL PRESSURE COEFFICIENT C_{pt} *vs.* ADVANCE DIAMETER RATIO V/nD FOR *four-bladed* PROPELLER FOR VARIOUS BLADE ANGLES. TOTAL EFFICIENCIES SHOWN IN CONTOUR

for given values of K. Several such lines are plotted on these charts for various values of K.

As a first step in the selection of a propeller to be used as a fan in a non-recirculating system, a tentative diameter can be chosen. Assuming that the flow rate is known, the L/D ratio for the system can be estimated by conventional methods of calculating friction loss in ducts. An average value of f, such as 0.02, may be assumed or if more precision is desired f may be evaluated taking into account the Reynolds number of flow and the inside surface condition of the duct system. In any event, once L/D and f are decided upon, the value of K is fixed. This value of K can be followed along the family of constant K lines on Figs. 7, 8 or 9 to the point of maximum efficiency. The point thus determined will fix the values of V/nD, C_{pt} and blade angle. The fan speed n

may be calculated from the V/nD value since with the diameter previously decided upon the average velocity of flow V through the fan would be fixed for the given rate of air flow. With this value of n, then, and the values of C_{pt} and total efficiency read from the chart, the shaft power may be calculated from Equation 12. At this point it can be determined whether two, three, or four blades give the most advantageous values of fan speed and power required for the tentative propeller diameter selected. This procedure may be repeated for other assumed propeller diameters so that the most practical selection may be made.

For the purpose of clarifying this procedure, an example will be given.

Example 1: Suppose it is required that a fan supply 100,000 cfm of air at 70 F and a barometric pressure of 29.92 in. of mercury to a duct system of certain specified proportions and length. The system is to be non-recirculating.

Solution: A propeller diameter of 8 ft will be tentatively chosen. On this basis suppose the equivalent length of the duct system is equal to 50 fan diameters. This means that the friction loss through the duct system has been determined to be the same as the loss through a straight pipe 8 ft in diameter and 50 diameters long or 400 ft in this case. Therefore, $L/D = 50$. Assuming that the friction factor is $f = 0.02$, the value of the pressure drop coefficient from Equation 14 is $K = 1 + 0.02\ (50) = 2.0$.

At this point it is not known whether to use a two-, three-, or four-bladed propeller, but a two-bladed propeller will be tentatively selected. From Fig. 7. it is estimated that the maximum total efficiency attainable for $K = 2$ is 70 percent. This occurs for a value of $V/nD = 0.30$, $C_{pt} = 0.09$, and $\beta = 14$ deg. Since 100,000 cfm must pass through the 8 ft diameter fan, the average velocity through the fan must be $V = 4\ (100{,}000)/\pi(8)^2(60) = 33.2$ fps. Since $V/nD = 0.30$, $n = V/0.3D = 33.2/(0.3)\ (8) = 13.8$ rps.

The mass density of air is the weight density divided by the acceleration of gravity. For the stated condition of temperature and barometric pressure the weight density of air is 0.075 lb per cu ft and therefore the mass density is $\rho = 0.075/32.2 = 0.00233$ slugs per cu ft. The shaft power may be calculated by using Equation 12 which gives $P = (\pi/4)\ (0.09/0.70)\ (0.3)\ (0.00233)\ (13.8)^3\ (8)^5 = 6070$ ft-lb per sec or a horsepower of $6070/550 = 11.0$ hp. If the rotating speed of 13.8 rps is satisfactory for the power drive available for this example, further trial selections need not be made.

The rating point selected on the curve for $K = 2$, however, could be moved up or down on the curve without appreciably changing the efficiency. The location of the point selected would affect the fan speed as just calculated. If a speed appreciably below 13.8 rps were desired, the preceding calculation could be repeated using the curves for three or four blades. If this still did not yield a desirable speed it would be necessary to repeat these calculations using a different fan diameter.

In many cases for fan selection the air capacity in cubic feet per minute and the static pressure in inches of water against which this air must be moved are the initially known quantities. The conversion factor to convert pressure H in inches of water to pressure p in pounds per square foot is given by the following equation:

$$p = 5.2\,H \qquad (16)$$

If d is used to denote air density in pounds per cubic foot and g the acceleration of gravity in feet per (second) (second),
then

$$\rho = d/g \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (17)$$

The quantity $\rho V^2/2$ appearing in Equation 13 becomes

$$\frac{\rho V^2}{2} = \frac{d V^2}{2g} = p_v \text{ lb per sq ft} \quad \ldots\ldots\ldots\ldots \quad (18)$$

Now if Equation 13 is divided by the factor 5.2 it becomes

$$\frac{p_t}{5.2} = \frac{p_v}{5.2} + f \frac{L}{D} \left(\frac{p_v}{5.2} \right) \quad \ldots\ldots\ldots\ldots \quad (19)$$

which reduces to

$$H_t = H_v + f \frac{L}{D} H_v \quad \ldots\ldots\ldots\ldots\ldots \quad (20)$$

where H_t and H_v are now total and velocity pressures respectively in inches of water. Since by definition $H_t = H_v + H_s$ where H_s is static pressure in inches of water, it follows from Equation 20 that

$$H_s = f \frac{L}{D} H_v \quad \ldots\ldots\ldots\ldots\ldots \quad (21)$$

or rearranging

$$f \frac{L}{D} = \frac{H_s}{H_v} = \frac{(2)\ (5.2)\ H_s\ g}{V^2 d} = \frac{335\ H_s}{V^2 d} \quad \ldots\ldots\ldots \quad (22)$$

In this equation V is the velocity through the fan in feet per second and its value depends on the fan diameter chosen. In the selection of a fan, however, a tentative diameter may be selected and the value of V computed corresponding to the required fan capacity. Then for the required static pressure the quantity $f(L/D)$ may be obtained from Equation 22 and substituted in Equation 14. From this point the calculations would proceed as illustrated in the example presented previously.

If a fan is installed in a duct system in which the air is recirculated through the fan, the fan is required to overcome only the friction loss in the system since the kinetic energy of flow is not dissipated. For this case Equation 13 reduces to

$$p_s = f \frac{L}{D} \frac{\rho V^2}{2} = K \frac{\rho V^2}{2} \quad \ldots\ldots\ldots\ldots \quad (23)$$

where

p_s = static pressure loss in the system.

K, the friction drop coefficient, is now defined as

$$K = f \frac{L}{D} \quad \ldots\ldots\ldots\ldots\ldots \quad (24)$$

This definition of K is applied to Figs. 4 to 6 inclusive. If the static pressure in inches of water against which the fan must operate is known, the value of

$f(L/D)$ may be calculated from Equation 22. This also is the value of K for the recirculating fan as is evident from Equation 24.

The method of determining the number of propeller blades, the blade angle, the propeller rotating speed, and the shaft power for a given propeller diameter and rate of air flow for the case of the recirculating system is identical with that employed for the non-recirculating system. For the recirculating system, of course, K is defined by Equation 24 and Figs. 4 to 6 must be used.

The dimensionless equations presented in this paper are not commonly employed in fan engineering and it may be of benefit to multiply some of them by suitable constants so the terms used may be expressed in conventional units.

Equations 7 and 8 may be rewritten as

$$C_{ps} = \frac{603000\ H_s}{d(\text{rpm})^2\ D^2} \quad \ldots\ldots\ldots\ldots \quad (25)$$

$$C_{pt} = \frac{603000\ H_t}{d(\text{rpm})^2\ D^2} \quad \ldots\ldots\ldots\ldots \quad (26)$$

where

(rpm) = fan speed, revolutions per minute.
H_s = static pressure, inches of water.
H_t = total pressure, inches of water.
d = density of air, pounds per cubic foot.
D = fan diameter, feet.

Equation 12 may be expressed as

$$(\text{bhp}) = 2.62 \times 10^{-10} \left(\frac{C_{ps}}{\eta_s}\right) (\text{rpm})^3\ D^5 Qd \quad \ldots\ldots\ldots \quad (27)$$

if static pressure and static efficiency are used or as

$$(\text{bhp}) = 2.62 \times 10^{-10} \left(\frac{C_{pt}}{\eta_t}\right) (\text{rpm})^3\ D^5 Qd \quad \ldots\ldots\ldots \quad (28)$$

if total pressure and total efficiency are used. In these equations (bhp) is brake horsepower supplied to the fan and Q is the air flow through the fan in cubic feet per minute.

Another useful conversion equation is

$$\frac{V}{nD} = \frac{1.275\ Q}{(\text{rpm})\ D^3} \quad \ldots\ldots\ldots\ldots \quad (29)$$

where V is velocity through the fan in feet per second and n is fan rotating speed in revolutions per second. The other quantities are as defined in previous paragraphs.

Another example will now be worked out to indicate how the performance of a propeller fan geometrically similar to those tested may be predicted.

Example 2: Suppose a three-bladed propeller 9 ft in diameter running at 1000 rpm is required to deliver 114,000 cfm against a static pressure of 1.05 in. of water. The air delivered is to be recirculated through the fan. Atmospheric air density is 0.075 lb per cu ft.

Solution: From Equation 25, $C_{ps} = (603000)\ (1.05)/(0.075)\ (1000)^2\ (9)^2 = 0.104$. From Equation 29, $V/nD = 1.275\ (114000)/1000\ (9)^3 = 0.20$. From Fig. 5 using these values of C_{ps} and V/nD, it can be seen that the blade angle must be 11 deg

and the static efficiency will be 55 percent. Finally, using Equation 27, the brake horsepower is computed to be bhp $= 2.62 \times 10^{-10}$ (0.104/0.55) $(1000)^2$ $(9)^2$ (114000) (0.075) = 34.4.

CONCLUSIONS

The data presented in this paper are applicable to two-, three-, and four-bladed propellers having the geometrical shape defined in Fig. 2 when operating as axial flow fans. These data are limited to the case of a propeller mounted in a round duct without hub fairing or guide vanes.

Examples have been worked out showing how a propeller may be selected for a specified air capacity and pressure rise. It was also shown how the blade angle setting and the horsepower required for a specific set of operating conditions could be determined.

The results of these model tests can be expected to be applicable to prototype fans so long as the tip speeds do not exceed $39\sqrt{T}$ fps where T is the air temperature in Rankine degrees. This corresponds to a tip Mach number of 0.8. In making this statement it is assumed that the effect of variation of the Reynolds number of flow around the propeller blades has a negligible effect on fan performance. It is believed that this is a safe assumption, particularly in the region of operation near maximum fan efficiency.

It is suggested that the methods used in this investigation might be employed whenever the performance of any fan might be required and it is not feasible to test the prototype. Tests could be conducted on a geometrically similar model and the results presented in the form given in this paper. This would serve as a basis for predicting the performance of the prototype fan.

DISCUSSION

A. E. CRIQUI, Buffalo, N. Y. (WRITTEN): The authors have demonstrated a useful method of plotting fan performance data in dimensionless form. They are wise in showing the method for use both for static pressure and for total pressure.

Especially for low pressure coefficient fans such as they discuss, the difference between these two pressures (which is the velocity pressure) is an important factor. The velocity should fit the needs of the installation. If fans of different outlet size (therefore different velocity) are to be compared, the performance which results after transforming the outlet area to the duct area should be used.

These problems do not appear in the paper, because the fan size and the duct size are always identical.

For the second determination of a fan size it is useful to use the relation of the pressure and velocity coefficients to diameter.

These are C_{pt} varies as $\frac{1}{D^2}$, $\frac{V}{nD}$ varies as $\frac{1}{D^3}$

The effect of size change could thus be plotted across the chart and be helpful in determining a new diameter.

AUTHORS' CLOSURE (E. C. LUNDQUIST): The pressure drop coefficient K given in Equations 14 or 24 defines the frictional characteristics of the duct system in terms of the equivalent length diameter ratio of the system based on the fan diameter. In

case the fan outlet is of a diameter different than that of the fan itself, the equivalent value of $f(L/D)$ can still be determined, since from Equation 22, $f(L/D) = H_s/H_v$. In this case, of course, it must be understood that H_v is the average velocity pressure at the fan section itself, and not at the fan outlet; and H_s is the static pressure rise required across the fan stage only. Thus, any losses entailed in the transformation from the fan diameter to the fan outlet diameter are included in H_s as part of the total static pressure losses in the duct system.

In answer to Mr. Criqui's comment on this point, the investigation described in this paper was not extended to cover the cases for outlet diameters different than the fan diameter. Curves similar to those plotted in Figs. 4 to 9, based on the velocity at the fan outlet, and with the value of K in Equations 14 and 24 based on the fan outlet diameter, could be reconstructed for various ratios of fan outlet to fan diameter if suitable assumptions regarding the losses in the diffuser sections could be made. In the absence of test data for various ratios of fan outlet diameter to fan diameter however, it is believed that satisfactory results can be obtained by using Figs. 4 to 9 just as they appear and considering the outlet diffuser on the fan as part of the external duct system.

1423

No. 1423

ROOF SPRAY FOR REDUCTION IN TRANSMITTED SOLAR RADIATION

By G. E. SUTTON,* GAINESVILLE, FLA.

THE UTILIZATION of solar energy[1] has been given much study at the University of Florida and other institutions. In Florida the solar radiation incident on a horizontal surface may be as much as 2300 Btu per (sq ft of surface) (day). This is nearly the heat equivalent of one horsepower-hour, or about two-thirds of the equivalent of one kilowatt-hour. If the heat received is visualized for an area of 1000 sq ft, the value of utilizing this energy is obvious.

There are cases, however, where this energy produces undesirable results. For instance, 320 Btu per sq ft per hr incident energy on an asphalt-slag roof will produce a surface temperature of about 150 F. It is obvious that, with an inside air temperature of 90 F or less, considerable heat transfer would take place, especially if the roof were of light construction.

This heat in conjunction with heat transfer through walls, windows, and other surfaces may serve to increase the temperature within the space to undesirably high values. In the case of warehouses, certain goods may deteriorate at such high temperatures. In airconditioned buildings, this heat may form a substantial part of the cooling load.

Any means which reduces the surface temperature will reduce the transfer of heat into the space correspondingly. One of the means used has been the placing of permanent pools of water on the roof[2-5]. Date storage buildings in California have been built with direct expansion or cold water coils on the roof. In some cases, refrigeration coils have been placed just under the roof but in such installations the difference in temperature between the top and bottom of the roof creates serious thermal stresses. While insulation serves to reduce the heat flow by increasing thermal resistance, rather than reducing the surface temperature, it causes higher surface temperatures which have a deteriorating effect upon the durability of asphalt roofing, due to such effects as swelling, blistering, and vaporization of the volatile oils[4]. Upon the occurrence of thunderstorms, quite

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[1] Exponent numerals refer to References.

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prevalent in Florida, the sudden cooling of the roof creates a serious thermal shock.

Experimenters at the University of Texas[2] have found that pools of water two inches and six inches in depth would maintain a surface temperature of 108 F and 103 F respectively. One disadvantage of waterpools is the structural difficulties involved in supporting the water, especially on large, flat buildings. The large quantity of heat stored in the water is also a disadvantage since a considerable part of it is transferred into the building in the late evening. Stagnant pools of water also present ideal locations for mosquito breeding, and for growth of algae[5]. The simplicity of pool systems is also reduced by the need

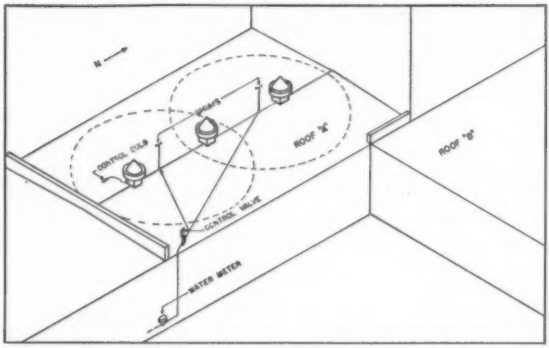


FIG. 1. ISOMETRIC VIEW OF ROOFS AND ROOF SPRAY INSTALLATION

for changing the water at regular intervals or for chemical treatment of the water.

The object of the experiment covered by this report was to determine the feasibility of using a roof spray as an economical means of reducing transmitted solar energy, and to determine factors affecting the design of a roof spray system.

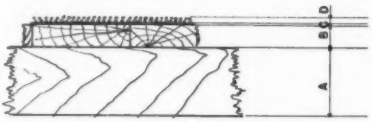
APPARATUS

Fig. 1 is an isometric sketch of the building and the spray installation. The two roofs designated by A and B were of identical construction of the type shown in cross section in Fig. 2. The two sections of the building house two portions of a machine shop, and are connected by a doorway. The slopes of the roofs are about one-half inch per foot, so slight that the roofs may be considered flat without serious error. Fig. 3 shows the sprays in operation.

The sprays were originally of the four-nozzle rotating-lawn type, having fixed nozzles of four different types for uniform coverage. These sprays were replaced with the adjustable two-nozzle rotating type which could be adjusted for throw and fineness of spray. The throw was set at about 18 ft with medium spray. The approximate coverage is shown by the dotted circles on Fig. 1.

Control of the system was based upon a sub-surface temperature. The control valve shown is an electric motorized type controlled by a temperature controller with the bulb placed as shown in Fig. 1. This bulb is buried under about ⅛ in. of asphalt and approximately ½ in. of slag. The bulb was placed thus in order to prevent frequent cycling of the system. If it were placed in view of the sun, it would dry much more rapidly than the roof, and would start the spray system before the roof had again reached the control temperature. This would result in considerable wastage of water, and very little added cooling. A slight time lag was expected due to the resistance to heat flow of the asphalt, but more positive control was expected.

The water meter was of the dial type, commonly used in city water systems. It was calibrated and found to be accurate within less than one percent at the flows required by the system.



A- 8"X10" WOOD FRAMING
B- 2" WOOD SHEATHING (PINE)
C- 2 PLY ASPHALT ROLL ROOFING
D- ASPHALT AND SLAG (ABT. 1/2")

FIG. 2. CROSS SECTION OF ROOFS USED IN TESTS

A continuous record of incident solar energy was made possible by use of a single point (curve drawing) recorder.

Ten temperatures were measured and recorded by a ten-point temperature recorder as follows: Outside air dry bulb, outside air wet bulb, spray water, sprayed roof surface, unsprayed roof surface, ambient at control bulb, ceiling surface under sprayed roof, air at 6 and 12 in. below the ceiling under the sprayed roof and at the breathing line (five feet from floor).

The rate of heat flow through the two roofs, and the surface temperatures of each ceiling, were measured by heat meters. The meters consisted of plastic plates containing an iron-constantan thermocouple and a multi-junction thermopile. They were used in conjunction with a portable potentiometer. A selector switch arrangement was made in order to facilitate readings, since copper leads were used with the thermopiles, and iron and constantan leads with a cold junction were required for the thermocouple.

A continuous record of spray operation was obtained from a rebuilt time-temperature recorder in which the recording pen was actuated by a solenoid, rather than a pressure-temperature element. A mercury switch installed on the actuating arm of the control valve operated the solenoid, recording the time when the sprays were turned on and off.

PROCEDURE AND RESULTS

Control Temperature. Control of the system was based upon a sub-surface temperature as previously stated. Data were collected at control settings of 95,

100 and 105 F. Results obtained at 95 F indicated that attempts to maintain any lower temperatures would be useless on clear days. Inclement weather was encountered before any temperatures higher than 105 F could be investigated.

As in most phases of engineering design, maximum conditions in this case should be considered. These would occur when there was no cloudiness, and on a day during the months of maximum solar radiation, June, July, August, and early September. The occurrence of an absolutely clear day during summer in Florida appears highly unlikely. However, the data collected on the days which approached nearest to this condition during each run were used in constructing Figs. 4, 5, and 6. August 8, 1949, was the nearest to an absolutely clear day encountered. For this reason heat flow and other data were obtained until the heat flow through the unsprayed roof returned to a value of zero.

Study of this heat transfer curve from a mathematical viewpoint shows the curve to resemble one wave of the sine curve (a plot of the equation $Y = k \sin X$). An empirical equation which very nearly fits the curve (disregarding the dip occurring at about 14:00†), is as follows:

$$q = 18.3 \sin \left[\frac{\pi}{12} (t - 9.5) \right] \quad \ldots \ldots \ldots \ldots \quad (1)$$

where:

q = heat flow, Btu per (hour) (square foot).
t = time (24 hour system), hours.

Fig. 7 shows a comparison of the curve obtained by this equation and the curve plotted from test data. Further use of this equation (modified) is discussed in the Appendix.

The heat transfer curve will be symmetrical about a time axis at 15:30. Observation will show that the curves of heat transfer through both sprayed and dry roofs are approximately symmetrical at this same time. Analysis of the sine curve will show half the area to be on each side of this vertical line. Integration of the observed curves to a time of 15:30 and multiplication by 2 should give the total heat flow for a particular day.

By using these considerations, the various days may be compared. The summary of the data from Figs. 4, 5, and 6 is given in Table 1, the heat flow being derived in the manner outlined.

The ratio of heat flow through the *dry roof* to that through the *sprayed roof* for control temperatures of 95, 100 and 105 F was 4.09, 6.3 and 3.98, respectively.

TABLE 1—SUMMARY OF TEST DATA FROM FIGS. 4, 5 AND 6

FIGURE NO.	TEMP. CONTROL SETTING F	TOTAL DAILY INCIDENT SOLAR RADIATION BTU/SQ FT	DAILY INWARD HEAT FLOW BTU/SQ FT		WATER USED GAL/SQ FT
			SPRAYED ROOF	UNSPRAYED ROOF	
4	95	2162	34	139.2	0.919
5	100	2180	22	138.8	0.485
6	105	1842	30	119.4	0.141

† 24 hr time system. This system was used to great advantage and convenience by the armed forces during World War II, during which the author became quite familiar with it. It is deemed much more convenient, especially in mathematical relationships, than the 12-hr system. The time used was Eastern Standard Time. Sun time is approximately 34 min earlier than Eastern Standard during July. This varies from 44 to 12 min, the maximum occurring during February, and the minimum during October.

A comparison of these data show an apparently lower efficiency at the 95 F control temperature. This will be explained under the discussion of *Water Requirements.*

Effect of Buried Control Bulb. A time lag of about 20 to 30 min was observed due to the covering of the control bulb with materials resistant to heat flow. This was noticed only on the first operation of the day. (See Figs. 4, 5, and 6, observing the initial rise of the sprayed surface to a temperature higher than the control setting.) During the rest of the day a difference in temperature

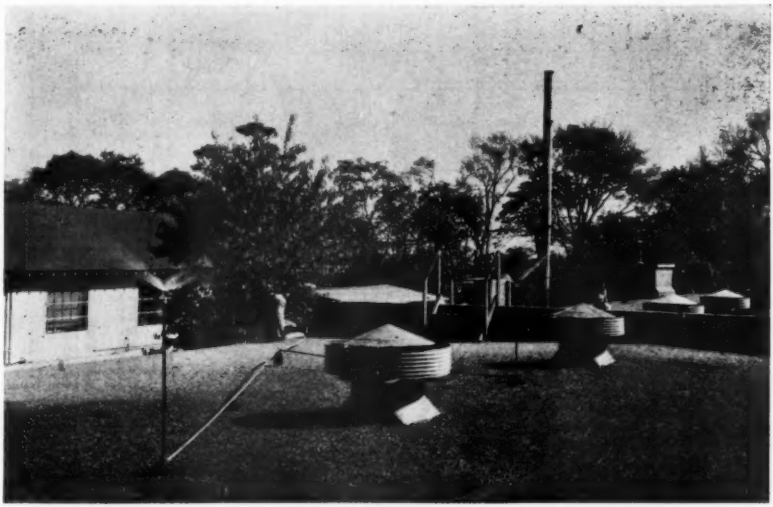


FIG. 3. VIEW OF ROOF SPRAYS IN OPERATION

between the surface and that of the material adjacent to the control bulb of more than two degrees was seldom encountered.

Water Requirements. An attempt was made to evaluate the relation between total solar radiation incidence and the amount of water used. A plot of this relation gives a wide variation of values, the correct curve for which is difficult to determine and consequently any plot of this curve would prove unreliable.

It may be observed from Figs. 4, 5, and 6 that the value of incident solar energy required to produce the various control temperatures appear to be as follows: 95 F = 190 Btu per (hr) (sq ft); 100 F = 200 Btu per (hr) (sq ft); and 105 F = 210 Btu per (hr) (sq ft). Mathematical analysis will show, however, that when the air temperature is 93 F, only slightly over one Btu per (hr) (sq ft) is required to produce a surface temperature of 95 F; and only about three Btu per (hr) (sq ft) are required to produce a surface temperature of 100 F. The amount of solar radiation occurring during the morning will then affect the rapidity with which the roof reaches a given temperature and will directly affect the amount of water consumption. The amount of solar radiation

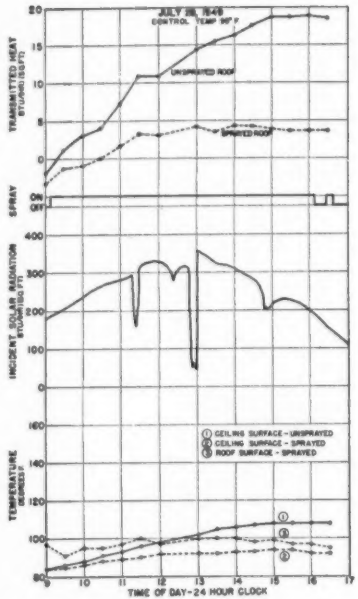


FIG. 4. RESULTS FOR BEST DAY AT 95 F CONTROL TEMPERATURE

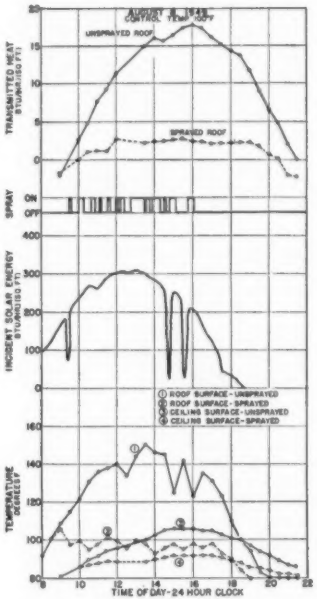


FIG. 5. RESULTS FOR BEST DAY AT 100 F CONTROL TEMPERATURE

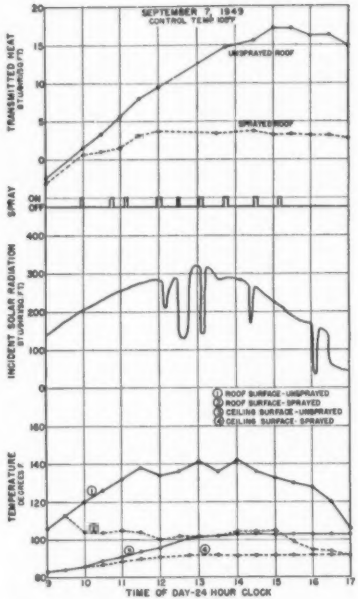


FIG. 6. RESULTS FOR BEST DAY AT 105 F CONTROL TEMPERATURE

during the rest of the day will govern the amount of heat to be dissipated. It may be concluded that, the nearer the solar radiation curve approaches the maximum curve, the more water will be required.

Insufficient data are available on solar radiation and cloudiness to determine whether or not the 1949 summer season was an average season. Therefore, no comparisons may be drawn for estimated water consumption on that basis. In

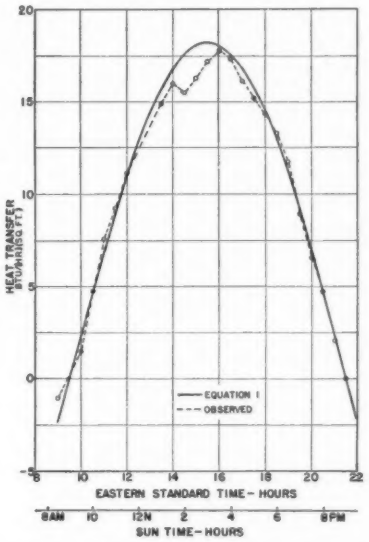


FIG. 7. COMPARISON OF OBSERVED VALUES OF HEAT TRANSFER WITH VALUES CALCULATED FROM EQUATION 1

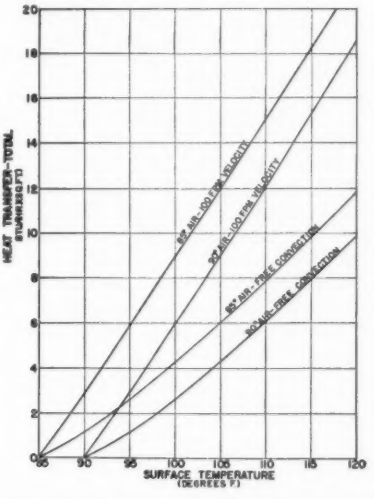

HEAT TRANSFER-TOTAL (BTU/HR/SQ FT)
95° AIR - 100 FPM VELOCITY
90° AIR - 100 FPM VELOCITY
95° AIR - FREE CONVECTION
90° AIR - FREE CONVECTION
SURFACE TEMPERATURE (DEGREES F)

FIG. 8. COMBINED HEAT TRANSFER BY CONVECTION AND RADIATION FROM PAINTED SURFACES TO AIR

this installation, the 2400 sq ft of roof required 85,767 gal of water between May 20, 1949, and September 30, 1949, or about 35.7 gal per sq ft.

Assuming that the evaporation of the water dissipated all the incident solar energy, the efficiencies (the ratio of this solar energy to the energy required for evaporation of the water) of the three runs based on all the data are 25.5, 48.9 and nearly 100 percent at control temperatures of 95, 100 and 105 F respectively. During the operation at the various temperatures differing spray conditions were observed. During the 95 F run considerable water remained pooled on the roof, and considerable runoff was observed. This would appear to show that most of the reduction in heat was due to warming of the water, and not to evaporation. During the 100 F run a slight amount of water ran off, and the roof remained only quite damp. During the 105 run, the roof remained fairly damp, and the runoff was extremely slight, just sufficient to show an abundance of water being supplied. These and other data, lead to the conclusion that the coverage of spray

was too heavy.[4-7] A somewhat finer spray has been recommended, but should not be so fine as to be a mist. A mist would allow too much evaporation in the air prior to impinging on the roof. A high wind drift loss would occur thereby decreasing efficiency and creating a possible nuisance. Finer sprays would require less water for operation and greater efficiency should be expected.

If each of the 150 days between May 1 and September 28 were to be clear, with an average incident solar energy of 2200 Btu per (sq ft) (day), 40 gal

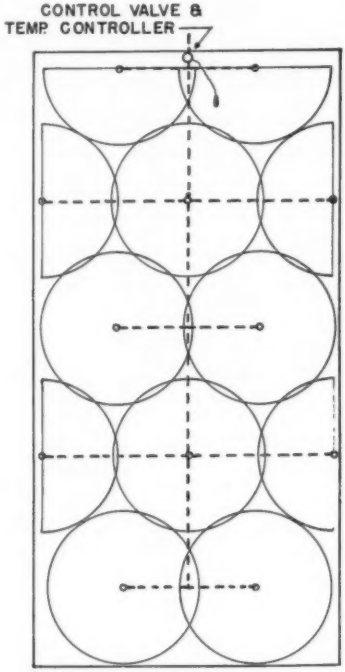


FIG. 9. LAYOUT OF SPRAY SYSTEM FOR MEDIUM-SIZED ROOF USING HALF- AND FULL-CIRCLE SPRAYS

of water evaporated at 100 percent efficiency would suffice to dissipate all of the solar energy for one square foot. Since the efficiency does not reach 100 percent, and since, conversely, each day is not clear, the conclusion is that the value of 40 gal per (sq ft) (season) is sufficiently high for a safe design figure.

Effect on Temperatures in the Space. The two roofs used for comparison were adjacent portions of a machine shop, connected by a doorway. Both spaces were well ventilated during working hours. The average air velocity adjacent to the ceiling was 100 fpm.

A decided cooling effect was noticed on the air above the roof, with some of this air spilling over the edges. During the time the sprays were in operation, and even with the roof wet, a decrease in air temperature of two to five degrees was recorded by the dry bulb thermometer in the psychrometer placed on the west wall of the sprayed building. This cooled air was utilized to some extent by the ventilation of the spaces.

No difference was noticeable at any time in the wet bulb temperatures in the two spaces. During the time when the spaces were well ventilated, no difference was noticeable in the dry bulb temperature. Observations on a day when the space was closed showed a temperature difference of slightly more than one degree Fahrenheit in the dry bulb temperature. Despite the low temperature differential between the two spaces a considerable difference in comfort existed. On a day when the air temperature was 93 F dry bulb and 78 F wet bulb, and the space ventilated, the space under the sprayed roof was reasonably comfortable. In the other space, exposed parts of the body such as face and arms felt decidedly hot and uncomfortable in spite of the air movement. The effective temperature (ET) for a dry bulb temperature of 93 F and a wet bulb of 78 F, with an air movement of 100 fpm, is 83 ET.[8] The elevation of effective temperature for an increase in mean radiant temperature is approximately in the ratio of one to three.[8] With a ceiling surface of 108 F, the elevation in effective temperature would approach five degrees. An effective temperature of 88 ET with 100 fpm air velocity and 78 F wet bulb would be equivalent to a dry bulb temperature of about 112 F. Since the walls are at a lower temperature than the ceiling, the mean radiant temperature is actually somewhat lower than 108 F, but the parts of the body which can be *seen* by the ceiling will be subject to the radiant effect. In a closed space, the effective temperature is slightly higher due to the lesser air movement.

Fig. 8 shows the heat transfer from a painted surface (emissivity of 0.90) at various temperatures to air at 85 and 90 F for air movements of 100 fpm and for natural convection. This is combined radiation and convection. The amount of heat transferred by radiation is dependent only upon the temperature differential between the surface and the air. The convected heat depends upon both the temperature differential and upon the velocity of the air. Therefore, the heat transferred with the 100 fpm movement is considerably higher than with only natural convection.

DESIGN OF SPRAY INSTALLATIONS

The most important features of the spray installation are the sprays and the spray control. A wide variety of spray heads is available. Fixed sprays, with heads giving full-, three-quarter-, half- and quarter-circle coverage with from 7- to 18-ft radius, having a great variety of elevation and fineness, are available. These should require less maintenance than rotating type sprays, provided the system is safeguarded against fouling by the use of strainers. Fig. 9 shows a typical layout of a spray system for a flat roof, using full- and half-circle sprays.

Sloping or gabled roofs may, in some cases, be more easily sprayed by perforated pipes. Fig. 10 shows an installation utilizing this form of spray. In large installations of this type, especially where the orientation of the building

is such that the gable runs north and south, control of each slope separately and the use of two pipes at the crest may prove more economical.

For small installations (one or two sprays), the ordinary lawn spray of the rotating type will be easier and more economical to install. These, too, are available in a wide variety.

In any type of installation, the system can only operate economically if it is thermostatically controlled. Many varieties of required devices such as motor-

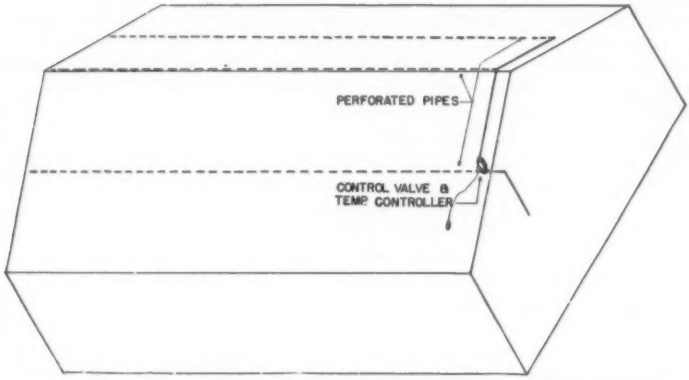


FIG. 10. LAYOUT OF SPRAY SYSTEM FOR PITCHED ROOF USING PERFORATED PIPES

ized valves, solenoid valves, and temperature controllers are available and can be arranged to best suit the individual installation. In most cases, electrical controls can be conveniently used or, where electricity is undesirable or unavailable, pneumatic controls may be used. For very large roofs zone control should be used. It is recommended that the piping be of brass or copper to prevent corrosion. This is especially important if any other than city water is to be used.

CONCLUSIONS

The following conclusions may be drawn from this investigation:

1. A roof spray of the rotating type can be made to maintain a surface temperature of about 100 F on a roof which would normally have a surface temperature of 150 F during the conditions of maximum incident solar energy.
2. The higher the surface temperature that may be tolerated, the more efficiently will the spray system operate, since more of the evaporative effect may be utilized.
3. Sprays are more effective than pooled water for cooling roofs.
4. Temperatures and comfort conditions within a space below a sprayed roof are desirably affected by spraying.
5. Some reduction in surface temperature of the walls of the building is produced by the spillage of cooled air from the roof.
6. Spray water has a preservative effect on asphalt types of roofing by reducing temperature change and lowering surface temperatures. There is, consequently, less thermal stress, less vaporization of volatile oils than for unsprayed roofs and there is no thermal shock from sudden thunderstorms.

7. Addition of insulation will decrease heat transmission, but will cause an increase in surface temperature, thus adding a deteriorating effect to some types of roofing.

8. Installation of sprays on conditioned buildings may serve to reduce the required refrigeration capacity, and thus present a saving in the cost of materials and installation.

9. The spray system, though simple in design, presents an efficient and economical means of reducing the effects of incident solar energy on roofs.

ACKNOWLEDGMENTS

The author wishes to acknowledge the valuable aid of many members of the staff of the Florida Engineering and Industrial Experiment Station, University of Florida, both in carrying out the experiment and in preparing the report.

Special aid was rendered by L. H. Holder, April Showers Co., Washington, D. C., in making available reprints of considerable reference material.

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APPENDIX

USE OF AN EMPIRICAL EQUATION TO PREDICT HEAT FLOW THROUGH VARIOUS TYPES OF ROOFS

Two characteristics of materials affect the transfer of heat through different materials subjected to the same temperature differential. These are thermal conductivity, or ability to conduct heat; and thermal diffusivity, or the ability to diffuse heat within the material itself. Both affect the amount of heat transferred and the

time lag between the time the temperature differential is applied and the time when heat actually begins to flow out of the material on the cooler side. In the case of periodic heat flow, that is, heat flow for conditions where the variations in temperature follow identical cycles the period of the cycle affects the amount of heat transferred. Three factors affect the heat transfer: (1) U, the overall coefficient of heat transfer, Btu per (hour) (square foot) (Farenheit degree); (2) λ, a dimensionless reduction factor which is the ratio of the actual heat flow to that which would occur under steady state conditions; and (3) dt or time lag, hours.

The reduction factor λ is calculated from the equation[9]:

$$\lambda = e^{-x\sqrt{\frac{\pi n}{\alpha}}} \quad \ldots\ldots\ldots\ldots \text{(A-1)}$$

where

λ = reduction factor.
e = base of Naperian logarithms, 2.7128 . . .
α = thermal diffusivity, square foot per hour.
n = number of periods per hour.
x = thickness of material, feet.

The thermal diffusivity, α, may be calculated from the equation[9]:

$$\alpha = \frac{k}{c_p \rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(A-2)}$$

where

k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
c_p = specific heat, Btu per (pound) (Fahrenheit degree).
ρ = density, pounds per cubic foot.

The time lag, dt, may be calculated from[9]:

$$dt = \frac{x}{2}\sqrt{\frac{1}{\alpha \pi n}} \quad \ldots\ldots\ldots\ldots \text{(A-3)}$$

where

dt = time lag in hours.
x = thickness of material, feet.

Using a weighted average of these values for the specific roof under consideration, Equation 1 of the paper may be revised to give the equation:

$$q_1 = 64.3\,\lambda U \operatorname{Sin}\left[(t\text{-}8\text{-}dt)\right] \quad \ldots\ldots\ldots\ldots \text{(A-4)}$$

where

q_1 = instantaneous value of heat flow, Btu per (hour) (square foot).
λ = reduction factor.
dt = time lag, hours.

This equation is limited to the hours between 8:00 and 22:00, since the temperature variation curve is of the sine form only during the hours of incident solar energy and a few hours thereafter.

Using Equation A-4 as a basis for calculations, Fig. A-1 shows the probable heat flow through three roofs: curve 1 is for a corrugated galvanized iron roof; curve 2 is for the roof used in this experiment, (see Fig. 2); curve 3 is for the roof as used in this experiment plus 2 in. of rock wool insulation. Curve 4 was added for comparison, and is the transfer through the test roof used, when it was sprayed to obtain 100 F surface temperature. Table A-1 shows three characteristics of the roofs.

These three types of roofs typify light, medium and heavy constructions. Equation

A-4 is approximate, but indicates the effect of increased resistance to heat flow, and increased time lag. Integration of the three curves of Fig. 9 shows a total heat flow of 22 Btu per (sq ft) (day) through the sprayed roof at a control temperature of 100 F.

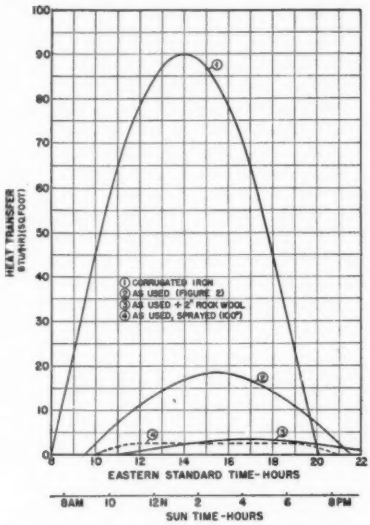

FIG. A-1. HEAT TRANSFER THROUGH THREE TYPICAL ROOFS AS CALCULATED FROM EQUATION A-4 AND THROUGH TEST ROOF CONTROLLED AT 100 F

TABLE A-1—CONDUCTIVITY, REDUCTION FACTOR AND TIME LAG FOR THREE ROOFS

ROOF TYPE	U BTU/(HR) (SQ FT) (F DEG)	λ	dl HOURS
Corrugated iron	1.40	1.00	0.00
As used in test	0.39	0.73	1.50
As used in test plus 2-in. rock-wool insulation	0.11	0.47	3.00

Though the insulated roof shows a heat flow of only slightly more than the sprayed roof, note that the heat flow continues far into the night; also the theoretical maximum surface temperature would be 175 F, or 25 deg higher than the dry roof without the insulation.

DISCUSSION

G. V. PARMELEE, Cleveland, O.: Mr. Sutton has presented a very interesting paper and might have indicated that he is also doing some interesting work on the subject of periodic heat flow through roofs.

I would like to ask Mr. Sutton to tell us of his experience in measuring solar radiation intensities, particularly with regard to the intensities which occurred on days of very high dry bulb temperature. In Cleveland we have found that on such days, which are also high dew point days, we have much less intense radiation than on clear cool days.

Secondly, I would like to ask if he knows of any work which is being done in studying the life of a roof as it is affected by insulation. Insulation will raise the roof temperature and probably cause some reduction in its life.

Thirdly, I would like to discuss the statement made in the paper regarding radiation loss from the roof. It was stated that the loss is dependent upon a difference in temperature between the roof surface and the outdoor air. It apparently was assumed that the sky was a black body radiator at air temperature. We have made a number of observations at Cleveland with a special radiometer of the radiation from the sky which falls on horizontal and other surfaces exclusive of solar radation. We find that the sky is not at all like a black body radiator at air temperature.

This radiation received from the sky is dependent upon air temperature and the dew point temperature and may vary from sixty to eighty percent of the radiation emitted by a black body at air temperature. We feel this is a rather significant matter in cooling and heating load calculations and are preparing a paper on the subject.

F. E. INCE, St. Louis, Mo.: I would like to question the author regarding the wind velocities present at the time of his test. Apparently that would affect the results of the evaporation. Secondly, has there been any reported windage loss of the spray?

A. S. HARON, Maywood, Ill.: I would like to ask if the author investigated the effect of relative humidity of the air on the cooling that one achieves with this spray.

AUTHOR'S CLOSURE: The first question by Mr. Parmelee was on the measurement of solar radiation. The maximum solar radiation occurs immediately after a thunderstorm when the air is clear. Concerning the effect of the dry bulb temperature, it is true that the maximum solar radiation does not occur simultaneously with the maximum dry bulb temperature.

There was no attempt to measure windage losses. A definite effort was made to control the spray so there would be no wind losses. They were low-angle sprays and the wind losses were kept down to a minimum. The velocities of the wind were very seldom more than 5 mph. When we have wind much more than that the temperatures and the solar radiation are not such that spray is required any great extent of the time. We were interested very intensely in the solar radiation; consequently, it was not studied from purely an air temperature viewpoint. Perhaps that should have been done.

Concerning Mr. Parmelee's second question about the effect of insulation on the life of roofs, we haven't done any work on that, as it is a long-range proposition. The particular roof we are using certainly appears to be in far better condition than those of equal age.

In answer to the comments of Mr. Ince, wind velocities were not measured; however, from previous measurements it would appear that most of the tests were conducted with wind velocities of from 3 to 5 mph. Winds of higher velocities were usually accompanied by cooler air requiring less water consumption and resulting in less wind losses. Wind losses are observable during operation when winds exceeded approximately 5 mph. The amount of wind losses depends upon the type of spray and the manner of coverage.

In answer to Mr. Haron's comments, the effect of relative humidity was not studied. The tests were performed throughout the summer, and the days were reasonably consistent as to humidity. A wet bulb temperature of 78 or 79 F occurred in a very large percentage of cases. Consequently, any study based on relative humidity would probably have been rather inconclusive.

1424

No. 1424

SEMI-ANNUAL MEETING, 1951

PORTLAND, OREGON

THE semi-annual meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held at Portland, Oregon, *City of Roses,* July 2nd, 3rd, and 4th, with 545 members, guests and ladies in attendance, establishing a new summer meeting record.

There were three technical sessions at which twelve papers on a variety of subjects were presented and discussed.

The first session was presided over by Pres. Lauren E. Seeley, who announced that developments during the year had indicated the need for revision of several sections of the By-Laws, and the Council had drafted and endorsed ten amendments in accordance with Article IX of the By-Laws.

H. E. Sproull, Chairman of the Charter and By-Laws Committee explained the proposed amendments which follow:

ARTICLE II—Memberships

Section 3. Qualifications

(d) Members. Graduation from an engineering school approved by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, shall be deemed equivalent to four (4) years of such experience; from other technical schools maintaining four (4) year courses, three (3) years; from a non-engineering college, two (2) years; and each successfully completed year in an engineering college shall be deemed equivalent to one (1) year of such experience.

To be amended as follows:

[Graduation from an engineering school with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, or from an engineering school of collegiate grade approved by the Council,] . . . of such experience.

(e) Junior Members. Any person over twenty (20) and under thirty-three (33) years of age, having graduated from a college or school of engineering approved by the Engineers' Council for Professional Development, or from a foreign college or school of engineering maintaining similar standards, or possessing five (5) years of experience satisfactory to the Council in the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, shall be eligible to become a Junior Member. Each successfully completed year in such college or school shall be deemed equivalent to one (1) year of such experience.

To be amended as follows:

(e) Junior Members. [Any person over twenty (20) and under thirty-three (33) years of age, having graduated from a college or school of engineering with an engi-

neering curriculum accredited by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, or from an engineering school of collegiate grade approved by the Council.] . . . of such experience.

(h) Students. Any person regularly enrolled and pursuing an engineering curriculum accredited by the Engineers' Council for Professional Development in a college or school of engineering, or in a foreign college or school of engineering maintaining similar standards, shall be eligible for student membership.

To be amended as follows:

(h) Students. Any person regularly enrolled and pursuing an engineering curriculum accredited by the Engineers' Council for Professional Development in a college or school of engineering, or in a foreign college or school of engineering maintaining similar standards, [or in an engineering school of collegiate grade approved by the Council,] shall be eligible for Student membership.

Section 4. Privileges and Limitations. (Student membership shall cease at the end of the calendar year in which the Student's enrollment as a student has been terminated.) The Council may transfer a Student to Junior Member grade on proof of his qualifications therefor or to Affiliate grade.

Amend the sentence shown in parenthesis above as follows:

[Student membership shall cease upon the Student's graduation, or at the close of the school term during which his enrollment as a student has terminated.]

Section 7. Resignation. Any member may resign his membership by writing to the Executive Secretary, who shall present the resignation to the Council at its next meeting. If the dues of the resigning member (pro-rated on a quarter-annual basis) shall have been paid to the date of the receipt of the resignation, his resignation shall be accepted.

Second sentence to be amended as follows:

[If the dues of the resigning member shall have been paid to the end of the quarter-annual period immediately preceding the date of the receipt of resignation, his resignation shall be accepted.]

Section 8. Suspension and Expulsion.

(a) Non-Payment of Dues: If any member shall fail to pay his dues by April 1st, he shall be classed as delinquent and not entitled to vote; if such dues are not paid by July 1st, he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by December 1st, the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by December 31st, he shall cease to be a member of the Society, and upon his failure to cure such default by December 31st, his membership in the Society shall cease; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Council may, in its discretion, rescind any forfeiture of membership.

To be amended as follows:

(a) Non-Payment of Dues: If any [Member, Associate Member, Junior Member, or Affiliate] shall fail to pay his dues by April 1st, . . . by December 31st, his membership in the Society shall cease; [if any Student shall fail to pay his dues by December 31st, the delinquent Student's membership shall cease and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased;] provided that upon written application satisfactory explaining a default, accompanied by payment of dues, the Council may, in its discretion, rescind any forfeiture of membership.

ARTICLE IV—Funds

Section 3. Dues. Honorary Members, Presidential Members and Life Members shall be exempt from the payment of dues.

Unless changed by the Society at an Annual or Special Meeting, the annual dues of Members, Associate Members and Affiliates shall be twenty-five dollars ($25.00) and of Students, three dollars ($3.00). The annual dues of Junior Members shall remain at ten dollars ($10.00) until January 1, 1951, whereupon the annual dues of Junior Members shall be fifteen dollars ($15.00).

Annual dues shall become due and payable in United States currency, or its equivalent, in advance on January 1st of each year, and shall in no case be subject to refund. Dues of new and advanced members, except Students, shall be pro-rated on a quarter-annual basis, and shall be payable on the first day of the month following notification of election or advancement, and if not paid within three (3) calendar months after such notification, the election or advancement shall be automatically rescinded. The Council in its discretion may suspend payment of dues by members in the Armed Forces in time of war, or may in its discretion defer or remit the dues of any member for special cause. Future annual dues may be compounded at a three percent (3%) rate by payment to the Society of the worth of an annuity equal to the member's dues for the period for which dues would be required. Compounded payments shall in no event be subject to refund.

Rule 1. The payment of Students' dues is hereby deferred until October 1st of each year. If such dues are not paid by December 31st, the delinquent Student's membership shall cease, and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased. Students' dues shall henceforth be applicable to the annual period commencing October 1st and terminating September 30th. As of October 1st, 1950, and on October 1st of each year thereafter, the Executive Secretary shall bill all students for dues in accordance with this Rule. *(Adopted 10-22-50)*

Second paragraph to be amended as follows:

Insert period after ($25.00) in first sentence and revise 2nd sentence as follows: [The annual dues of the Junior Members shall be fifteen dollars ($15.00). The annual dues of Students shall be fixed by the Council and shall be published in THE JOURNAL.]

Third paragraph to be amended as follows:

Annual dues, [except Students,] shall become due and payable in United States currency, or its equivalent, in advance on January 1st of each year, and shall in no case be subject to refund. Dues of new and advanced members, [except Students,] shall be pro-rated on a quarter-annual basis, and shall be payable on the first day of the month following notification of election or advancement, and if not paid within three (3) calendar months after such notification, the election or advancement shall be automatically rescinded. [Students' dues shall be applicable to the annual period commencing October 1 and terminating September 30.] The Council in its, etc. . . .

Section 4. Dues to Constitute Subscriptions. Of the annual dues paid by Members, a sum equal to the current subscription price shall be deemed to be a subscription for the JOURNAL. All members, except those whose annual or pro-rated dues shall amount to less than five dollars and fifty cents ($5.50), shall be entitled to receive the Society's periodical publications. Honorary Members, Presidential Members and Life Members shall be entitled to receive such publications without charge. Students shall be entitled to receive the JOURNAL without charge.

Rule 1. The Executive Secretary shall have imprinted on the Society's bills for dues payable by all members including Students, the following legend: "Includes $2.00 for one year subscription for the monthly JOURNAL of the Society, HEATING, PIPING AND AIR CONDITIONING." *(Adopted 6-18-50)*

To be amended as follows:

Section 4. Dues to Constitute Subscriptions. [Of the annual dues paid by Members, Associate Members, Junior Members and Affiliates, a sum equal to the current subscription price shall be deemed to be a subscription for THE JOURNAL, and all such members, except those whose annual or pro-rated dues shall amount to less than Five and 50/100 dollars ($5.50), shall be entitled to receive the Society's periodical publications. Honorary Members, Presidential Members and Life Members shall be entitled to receive the Society's publications; the Society shall subscribe for THE JOURNAL in their behalf].

ARTICLE VII—Committees

Section 3. General Committees.

(c) Guide Committee, consisting of nine (9) MEMBERS. The said committee, in harmony with the editorial policies of the Society, shall compile the text section of THE GUIDE. The Director of Research shall be an ex-officio member of the said committee.

To be amended (2nd sentence) as follows:

[The Chairman of the Committee on Research shall be an ex-officio member of the said Committee.]

(f) Chapter Relations Committee, consisting of a member of the Council, who shall serve for a term of one (1) year, and six (6) MEMBERS. Each of the seven (7) regional areas, as set forth in Appendix A hereof, shall have representation on said Committee.

To be amended as follows:

(f) [Chapter Relations Committee, consisting of a member of the Council and six (6) MEMBERS, who shall serve for a term of one (1) year.] Each of the seven (7) regional areas, as set forth in Appendix A hereof . . ., etc.

(h) Committee on Research, (third paragraph):

The chairman of the Committee on Research shall appoint such *Technical Advisory Committees* as may be deemed expedient, to advise the Committee on Research and the Director of Research on specific projects. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research, and the chairman of the Committee on Research and the Director of Research shall be ex-officio members thereof. Technical Advisory Committees shall be governed by such rules and regulations as may be recommended by the Committee on Research and adopted by the Council.

To be amended as follows:

The chairman of the Committee on Research shall appoint such Technical Advisory Committees as may be deemed expedient to advise the Committee on Research and the Director of Research on specific research projects; [the chairman of the Committee on Research shall be an ex-officio member thereof.] At least one (1) member of each Technical Advisory Committee shall . . ., etc.

The following new sub-division to be added for the inclusion of the Public Relations Committee as a General Committee of the Society:

(i) Public Relations Committee, consisting of a member of Council and three (3) members who shall serve a term of one (1) year. The said committee shall publicize the aims, activities and achievements, the scientific, and educational purposes of the Society, with the object of cultivating and stimulating public and members' interest in the Society and its affairs.

A discussion followed the presentation of the proposed amendments, but no changes were suggested.

President Seeley appointed a Resolutions Committee as follows: Harry L. Stevens, Hutchinson, Kans.; C. M. Burnam, Jr., Chicago, Ill.; and R. T. Kern, Fitchburg, Mass. Three technical papers were presented by their authors, one by title, and I. W. Cotton, Chairman, gave a report on the activities of the Committee on Research.

At the second session held on Tuesday morning, first Vice-President Ernest Szekely presided and four papers were presented, one by title.

At the third and final session there were three papers presented by their authors, and one paper presented by title. Reg. F. Taylor, second vice-president, was the chairman at this session. The report of the Resolutions Committee was called for by President Seeley, who had assumed the chair, and was presented as follows:

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is concluding its highly successful and pleasant Semi-Annual Meeting 1951 in the gorgeous City of Roses of the Great Northwest, and

WHEREAS, the Oregon Chapter through Bert Farnes, the genial general chairman and his many busy beaver helpers, has kept us so enjoyably occupied that the Resolutions Committee has had no time to compose the elaborate and complimentary tributes that the Host Chapter properly deserves, and

WHEREAS, the Oregon Ladies have entertained our wives and daughters so charmingly, and

WHEREAS, the technical papers and discussions have been of a character and quality to reflect great honor and dignity to our Society, and to make an important contribution toward the comfort, health and long life of humanity, and

WHEREAS, President Lauren E. Seeley and the other Officers of the Society have conducted the technical sessions in an expeditious manner so that we might enjoy the culture, beauty and entertainment planned for us, and

WHEREAS, Portland has extended us such a hearty welcome, and

WHEREAS, our headquarters hotel, the Multnomah, has so successfully avoided *check in* and *check out* queues and reservation confusion, and has through the courteous attention of its staff and its many facilities provided for our every comfort, and

WHEREAS, our vocabularies have been so enriched by Multnomah, Mollala Buckaroo and many other romantic words not commonly encountered,

NOW, THEREFORE, BE IT RESOLVED THAT: We, the Members of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, formally, unanimously and vociferously extend a *Thank You* as hearty as our *Welcome to Portland*, to the president, committees and all of the members of the Oregon Chapter, to Portland, our host city, to the Hotel Multnomah, our headquarters, to the Royal Rosarians who honored President and Mrs. Seeley, to those contributing papers and discussions, to Lewis and Clark for discovering this great Northwest, to our own Society officers, committees and staff for a mighty fine meeting.

Respectfully submitted,

RESOLUTIONS COMMITTEE

HARRY L. STEVENS, *Chairman*
C. M. BURNAM, JR.
R. T. KERN

The meeting concluded with a rising vote adopting the resolutions of thanks. The final registration showed members present from 29 states, the District of Columbia, Alaska, Canada and Italy.

PROGRAM SEMI-ANNUAL MEETING

Hotel Multnomah, Portland, Oregon—July 2-4, 1951

SUNDAY—July 1

9:00 a.m. REGISTRATION (Hotel Multnomah, *Mezzanine*)

10:00 a.m. Council Meeting (Peekskill)

2:00 p.m. Sightseeing tour about City with Chapter Members in private cars

MONDAY—July 2

9:00 a.m. REGISTRATION (Hotel Multnomah, *Mezzanine*)

9:30 a.m. TECHNICAL SESSION (Hotel Multnomah, Marine Room)

Amendments to By-Laws

Aluminum Ceiling Panels for Heating and Cooling, by E. S. Howarth, S. C. Huddleston, and R. M. Koch, New Kensington, Pa., presented by Mr. Howarth

Laboratory Studies of the Thermal Characteristics of Plaster Panels, by C. M. Humphreys, C. V. Franks, and L. F. Schutrum, Cleveland, Ohio, presented by Cyril Tasker

Optimum Panel Surface Distribution Determined from Human Shape Factors, by F. W. Hutchinson, Berkeley, Calif., and Merl Baker, Lexington, Ky., presented by A. V. Hutchinson

Report of the Committee on Research, I. W. Cotton, Chairman

Barometric Draft-Control Analysis, by H. B. Nottage, D. W. Locklin, and R. G. Huebscher, Cleveland, Ohio, presented by title

11:00 a.m. Ladies Brunch (Hotel Multnomah, *Rose Bowl*)

11:30 p.m. Research and Eichberg Cups Golf Tournament, Glendoveer Course

12:30 p.m. Columbia River trip

7:00 p.m. Hawaiian Buffet Dinner Party (Hotel Multnomah, *Rose Bowl*)

TUESDAY—July 3

9:00 a.m. REGISTRATION (Hotel Multnomah, *Mezzanine*)

9:30 a.m. TECHNICAL SESSION (Hotel Multnomah, *Marine Room*)

Control of Air Streams from a Long Slot, by Alfred Koestel and Chia-Yung Young, Cleveland, Ohio, presented by Prof. G. L. Tuve

Pressure Losses of Take-Offs for Extended-Plenum Duct Systems, by J. W. Holl, S. F. Gilman, R. J. Martin, and S. Konzo, Urbana, Ill., presented by J. R. Jamieson

Warm Air Perimeter Heating—Heat Emitted from Floor Surface, by J. R. Jamieson, R. W. Roose, and S. Konzo, Urbana, Ill., presented by J. R. Jamieson

Furnace Thermal Efficiency Determined from Flue and Jacket Loss, by H. L. McPherson, K. L. Badger, and W. S. Zawada, Cleveland, Ohio, presented by title

10:00 a.m. Ladies Shopping Time

12:00 noon CITY OF ROSES LUNCHEON (Hotel Multnomah, *Rose Bowl*)

Toastmaster: Bert W. Farnes

Speaker: Dean G. W. Gleeson, School of Engineering and Industrial Arts, Oregon State College

Subject: Energy—Choose It Wisely Today for Safety Tomorrow

1:30 p.m. Nominating Committee (Hotel Multnomah, *Spruce Room*)
1:30 p.m. Chapters Conference Committee
2:30 p.m. Scenic Motor trip to Timberline Lodge and Mt. Hood
6:30 p.m. Indian-Style Salmon Bake at Mt. Hood

WEDNESDAY—July 4

9:00 a.m. REGISTRATION (Hotel Multnomah, *Mezzanine*)
9:30 a.m. TECHNICAL SESSION (Hotel Multnomah, *Marine Room*)
Heat Pump Application to a Newspaper Plant, by J. D. Kroeker, J. H. Bonebrake, and J. A. Melvin, Portland, Ore., presented by Mr. Kroeker
Two-Year Performance of a Heat Pump System Furnishing Year-Round Air Conditioning in Modern Office Building, by Philip Sporn and E. R. Ambrose, New York, N. Y., presented by Mr. Ambrose
Climatology as an Aid in Heat Pump Design, by G. S. Smith, Seattle, Wash., presented by Professor Smith
Heat Requirements of Buildings, by C. W. Signor, Detroit, Mich., presented by title
11:00 a.m. Mollala Buckaroo at Mollala
1:30 p.m. Inspections—Country's two largest heat pumps (Also available 10:00 a.m. July 5)
7:00 p.m. SEMI-ANNUAL BANQUET (Hotel Multnomah, *Grand Ballroom*)
Toastmaster: T. E. Taylor
Speaker: Frank Branch Riley
Subject: The Last West

COMMITTEE ON ARRANGEMENTS

B. W. FARNES, *General Chairman* F. F. URBAN, *Vice Chairman*

PROF. E. O. EASTWOOD, *Honorary Chairman*

Banquet—T. E. Taylor, *Chairman;* W. B. Hayes, *Vice Chairman;* E. A. Ponder, Mrs. Helen P. Woolley, and R. W. Zanders.

Entertainment—E. R. Lokey, *Chairman;* W. R. Norte, *Vice Chairman;* J. P. Bryne, J. E. Finkbeiner, D. M. Keefer, W. H. Oscanyan, W. R. Pindell, J. W. Wallace, Jr., and S. W. Young.

Finance—J. D. Kroeker, *Chairman;* L. J. Harrington, *Vice Chairman;* P. L. McManus.

Ladies—J. P. McDermott, *Chairman;* K. H. Hanson, *Vice Chairman;* J. H. Bonebrake, T. H. McClung, W. R. Norte, and Fred Schwan. Mmes. Dick Blankenship, J. H. Bonebrake, C. W. Brissenden, R. C. Chewning, B. W. Farnes, J. A. Freeman, K. H. Hanson, Walter Hanthorn, A. N. Hoss, J. D. Kroeker, T. C. Langdon, R. P. Lankow, E. R. Lokey, F. W. McCarthy, T. H. McClung, J. P. McDermott, H. W. McKenzie, W. B. Morrison, W. R. Norte, W. R. Pindell, Fred Schwan, W. A. Simpson, T. E. Taylor, B. M. Thornburg, and F. F. Urban.

Publicity—A. N. Hoss, *Chairman;* V. A. Beers, *Vice Chairman;* D. W. Chase, R. M. Hermann, E. R. Hoerner, W. J. Kollas, and F. E. Mulford.

Reception—W. B. Morrison, *Chairman;* O. W. Matthews, *Vice Chairman;* D. L. Benz, J. S. Bowen, Jr., W. R. Burton, C. P. Dougherty, R. E. Duke, R. M. Finklea, R. S. Freidenrich, R. O. Haneberg, H. H. Hewitt, Jr., F. R. Johnson, E. F. Kugel, R. F. Marquess, H. W. Olson, R. L. Schroeder, and S. W. Young.

Sessions—F. W. McCarthy, *Chairman;* W. C. Kelly, *Vice Chairman;* J. S. Bowen, Jr., Faruk Konuk, and W. D. Maxwell.

Sports—R. P. Lankow, *Chairman;* R. L. Schroeder, *Vice Chairman;* J. A. Reynolds, R. E. Selberg, and W. G. Woolley.

Attendance Promotion—C. W. Brissenden, *Chairman;* A. C. Clark, *Vice Chairman;* R. H. Brown, K. N. Flocke, and W. R. Pindell.

Transportation—J. A. Freeman, *Chairman;* W. A. Simpson, *Vice Chairman;* R. R. Atkinson, A. H. Bohren, J. T. Burtchaell, R. L. Carlson, E. E. Carroll, O. E. Castrow, L. D. Copenhagen, M. O. Cox, A. E. Finlay, W. A. Gish, Walter Hanthorn, L. O. Howlett, O. T. Jacobson, A. E. Kendrick, T. C. Langdon, A. L. Lewis, R. H. Lord, J. M. Lott, R. W. Martig, W. C. Miller, Benjamin Morris, E. A. Murhard, K. R. Murhard, H. H. Nesbitt, R. W. Peterson, V. A. Pfeifer, W. P. Ramsey, J. A. Reynolds, Jack Rousseau, R. B. Rowan, Henry Rudell, P. F. Sansted, E. P. Siegert, U. G. Smith, F. A. Staten, C. L. Walker, A. S. Weiant, W. J. Widmer, and W. G. Woolley.

Alaska Representative—R. D. Hoss.

Special Activities—Dick Blankenship, E. E. Kelly, R. T. Mudge, and B. M. Thornburg.

1425

No. 1425

ALUMINUM CEILING PANELS FOR HEATING AND COOLING

By E. S. HOWARTH*, S. C. HUDDLESTON**, AND R. M. KOCH**,
NEW KENSINGTON, PA.

DESIGNERS of modern air-conditioned buildings are confronted with the problem of providing, at reasonable cost, adequate heating, ventilating, and air conditioning facilities through the use of equipment and arrangements which will conserve the greatest amount of space for other purposes. In solving this problem in a 30-story office building being constructed for the Aluminum Company of America in Pittsburgh, Pa., investigation was made of a multi-purpose aluminum ceiling for heating, for partially cooling, and for accomplishing the desired acoustical treatment. In addition to providing the advantages of ceiling-panel heating, this installation will employ ceiling-panel cooling in a similar manner, thus relieving the air conditioning system of a large portion of its normal load. The resulting reduction in size of air handling equipment and ducts, the efficient use of ceiling area, and certain construction innovations adopted are expected to increase the net usable space by an amount equivalent to about one and one-half stories.

Aluminum ceiling panels offer certain advantages when compared with panels composed of water-carrying tubes embedded in plaster or concrete. The high thermal conductivity of aluminum permits wide spacing of the water-carrying tubes and obtains a temperature of the effective face of the panel approaching that of the water in the tube. The latter consideration makes panel cooling practicable because the temperature of the chilled water can be above the dew-point of the ambient air and still provide effective cooling. The aluminum panels respond rapidly to quick changes in water temperature and are not subject to the cracking hazards. Acoustical treatment is accomplished readily with no adverse effect on thermal performance by perforating the sheet elements of the panels and overlaying them with glass wool blankets in a manner common to metal acoustic ceilings.

A search of the literature revealed that considerable information concerning the design and thermal performance of radiant plaster panels was available from

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* Chief, Metal Working Division, Aluminum Research Laboratories, Aluminum Company of America.
** Research Engineer, Aluminum Research Laboratories, Aluminum Company of America.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

the work of Raber, Hutchinson, Vanderweil, Giesecke, and others. With the exception of a paper[1] by Leopold, dealing with the cooling performance of ceiling mounted aluminum plates having integral water passages, little technical information relating to metal ceilings for heating and cooling was found. Consequently, a program of tests was initiated to determine not only the heating and cooling performance of such ceilings, but also to indicate the relative effects of variations in design details and operating conditions upon thermal performance.

During the early stages of designing the Alcoa Office Building, consideration was given to the use of aluminum ceiling panels in which the water-carrying elements were an integral part of the ceiling or were either bonded or mechanically attached to it. Bonded construction, in which ceiling surface and water-carrying elements were brazed together, seemed to offer the best combination of fabrication, installation, and thermal characteristics. The test results reported in this paper were obtained from such brazed panels, but the analytical procedure is not in any way limited to that type of construction. In fact, the ceiling selected for the building is being constructed of relatively small aluminum pans having integral clips which attach the pans mechanically to the water-carrying pipes. The theory and design data presented in this paper have been applied to them with good results.

DESCRIPTION OF TESTS

The experimental phase of the investigation involved the testing of several designs of ceiling panels, each of which, in general, consisted of a sheet of aluminum to the top side of which was brazed aluminum tubing tracing a sinuous path across the panel. Most of the tests were performed on panels in which the facing sheet was ⅛ in. thick. However, tests were also conducted on panels having facing sheet thicknesses of 3/32 and ½ in. Five panel designs were tested in which adjacent tube passes were spaced on 8-in. centers and seven designs involved tubes spaced on 20-in. centers. The emissivity of the panel surfaces was varied by different surface treatments, namely, ivory paint (94 percent), anodic oxide coating (91 percent), lacquer (69 percent), aluminum paint (33 percent), and bare aluminum (4 percent). The thermal conductivity of the 3S aluminum facing sheets was determined by test to be 86 Btu per (hr) (sq ft) (F deg per ft). Each panel was 4 ft-3 in. wide by 7 ft-0 in. long and two such panels were assembled to make a continuous 14-ft long section. The aluminum tubing had a ⅝-in. outside diameter and a wall thickness of 0.065 in. The tests were conducted with an insulating layer on the top side of the panel consisting of a 1-in. thick blanket of glass wool covered on the top only by aluminum foil. Fig. 1 shows the top side of a typical panel, showing insulation and tubing. Only a small percentage of the panels tested were perforated.

The 14-ft long panel assemblies were tested, two at a time, while suspended from the ceiling of a specially constructed test room, one corner of which is shown in Fig. 2. The totally-enclosed test room was designed to represent the size of a typical office and measured approximately 12 ft by 16 ft by 8½ ft high. The walls, floor, and ceiling were insulated with thick rock wool batts. The

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[1] The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by Charles S. Leopold (*Refrigerating Engineering*, July 1947, p. 33).

finished floor was wood and the walls were faced with insulating wall board. Fluorescent fixtures, recessed flush with the ceiling panel surfaces, provided a lighting load of 3 watts per sq ft of floor area.

Nominal room air temperatures of 70 F for heating tests and 80 F for cooling tests were maintained by circulating air through the room. Air temperature control was obtained by employing external air conditioning equipment and pneumatic type temperature controllers. With this equipment the local variation in air temperature with time was held to within ±1 deg.

Water at any desired temperature was supplied to the panels by an external system of pumps and heat exchangers and water temperature variation was held

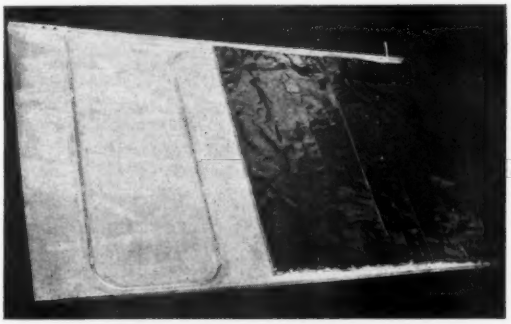

FIG. 1. TOP VIEW OF TYPICAL ALUMINUM CEILING PANEL—20-IN. TUBE SPACING

to within ±0.1 deg through the use of several pneumatic-type temperature controllers. Water flow rate was indicated by a calibrated flow measuring instrument. A general view of the external test room equipment, including air conditioning system, water temperature control system, and measuring instruments, is presented in Fig. 3.

Temperature measurements were made by means of calibrated iron-constantan thermocouples and a self-balancing electronic potentiometer employing a melting ice cold junction. Air temperature measuring thermocouples which could *see* the panels were shielded with aluminum foil to minimize direct radiation effects. Thermocouples were located at numerous points on the walls and floor of the test room and at ten different locations along the 14-ft centerline of each panel assembly. Other couples were suspended in the air at a number of points below and above the panels. Inlet, outlet, and intermediate water temperatures were measured. Each temperature survey involved approximately 60 measurements.

Air was circulated through the room at a rate of approximately 880 cfm, which was about equivalent to one change every 2 min. This rate was necessary to remove the heat released by the panels in order that the room air temperature would remain constant. Part of the tests were made with the inlet air grille louvers adjusted half up and half down. With this arrangement the air velocity across the face of the panel was measured with an anemometer at 30 locations on the two panels when cold, and the average velocity was found to be about

25 fpm. To obtain air conditions within the room that would more closely approximate the conditions set up by natural convection, a perforated diffuser partition was erected in front of the inlet air grilles. With the partition in place no indication of air motion anywhere in the room was obtained with the anemometer, although calibration data were available for velocities as low as 10 fpm. Therefore, it was concluded that for the tests made with this setup the air velocity across the surface of the panels was less than 10 fpm.

The interior corner of the test room, shown in Fig. 2, reveals portions of the ceiling panels with surface temperature measuring thermocouples attached, sev-

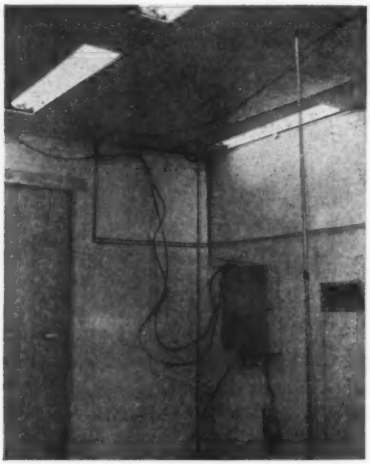

FIG. 2. INTERIOR VIEW OF ONE CORNER OF THE TEST ROOM

eral wall thermocouples, the standards supporting the air temperature measuring thermocouples, the thermocouple junction box, and one of the inlet air grilles. Fig. 4 is a view of this same corner of the test room with the perforated diffuser partition in place.

Tests were conducted employing inlet water temperatures of 180, 160, and 120 F for heating and 60 F for cooling. Water flow rates were established at either 1 or ¼ gpm. Test data were not recorded following a change in test conditions until equilibrium was reached and then three consecutive series of observations were made at about 10 min intervals.

RESULTS OF TESTS

The thermal performance of each panel under the various test conditions was calculated from the rate at which heat was released from, or absorbed by, the water as it flowed through the panel. Dividing this heat rate, Q, expressed in Btu per hour, by the room-side panel face area, A, in square feet, resulted in the average unit panel performance, Q/A, expressed as Btu per (hr) (sq ft). This index to panel performance includes the relatively small amount of heat transmitted through the insulating blanket on the top side of the panel.

A summary of the thermal performance results obtained from 89 heating tests and 30 cooling tests is presented in Table 1. The upper number in each block represents the average thermal performance, Q/A, obtained from three consecutive series of observations made under equilibrium conditions. From this summary, the effects of variations in panel design and operating conditions upon the thermal performance of the panel can be observed. It is important to note that improved thermal performance resulted from increasing face sheet thickness, surface emissivity, water flow rate, and air velocity, and from decreasing tube spacing. For the heating tests, raising the inlet water temperature resulted

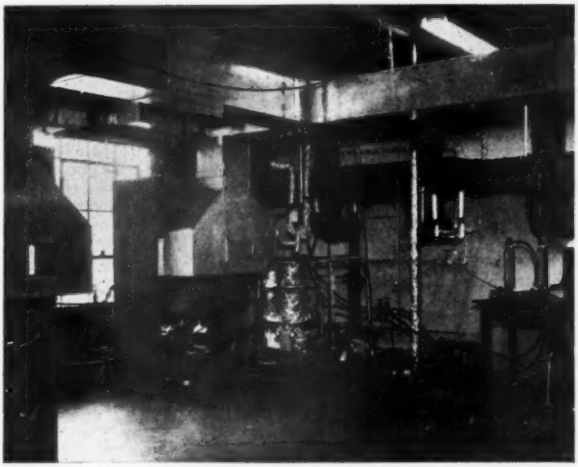

FIG. 3. EXTERNAL VIEW OF TEST ROOM, AIR CONDITIONING, AND WATER CONTROL EQUIPMENT

in a pronounced increase in thermal performance. Although not shown in Table 1, a similar improvement was observed for cooling tests in which the temperature of the inlet water was decreased below 60 F.

Interesting cross comparisons can also be made in Table 1. For example, it can be seen that, for the lower air velocity, introduction of 160 F water at 1 gpm resulted in almost the same heat output as was obtained for 180 F water at ¼ gpm. This suggests the use of water flow rate as well as water temperature as a means of controlling panel performance.

Further examination of the test results shown in Table 1 indicates that, for identical water flow rates, air velocities, face sheet thickness, and surface emissivities, panels having a tube spacing of 8 in. exhibited practically the same thermal performance for 160 F inlet water as did 20-in. tube space panels when the inlet water temperature was 180 F. Similarly, increasing the facing sheet thickness from ⅛ in. to ½ in. nearly compensated for an increase in tube spacing from 8 to 20 in.

In calculating the performance of the perforated panels, it was originally intended to consider only the net face area of the facing sheet. An analysis of the

NOMENCLATURE

W = width of panel, feet.
L = length of panel, feet.
A = room-side face area of panel, square feet.
b = thickness of panel facing sheet, feet.
D = developed length of tube, feet.
s = distance between centerlines of adjacent tube passes, (also called tube spacing), feet.
Sec. A-A = a typical cross section of the panel taken normal to the tube and extending in width a distance $s/2$ on each side of the tube.
x = shortest distance from any point on the face of the panel to the centerline of the tube projected to the face of the panel, feet.
y = distance from water inlet edge of the panel to Section A-A as measured along the tube, feet.
dy = small increment of y, feet.
$_{1, 2}$ = subscripts used to relate temperature to the inlet and outlet edges of the panel respectively.
Q = rate at which total heat is released from or absorbed by the entire panel, Btu per hour.
$q_{A\text{-}A}$ = rate at which total heat per unit area is released from or absorbed by the panel at Section A-A, Btu per (hour) (square foot).
G = water flow rate, gallons per minute.
w = water flow rate, pounds per hour.
p = specific heat of water, Btu per (pounds) (Fahrenheit degree).
k = thermal conductivity of facing sheet material, Btu per (hour) (square foot) (Fahrenheit degree per foot).
ϵ = emissivity of panel surface, percent.
e = Naperian constant, 2.718.
t_a = average temperature of room air, Fahrenheit.
t_r = weighted mean temperature of room air, walls, and floor, including space and structure above panels, (approximated by considering only the temperatures of the room walls and floor), Fahrenheit.
t_w = temperature of the water at any point a distance y, along the tube, Fahrenheit.
t = temperature of the facing sheet at any point, Fahrenheit.
t_t = temperature of the facing sheet at any point directly under the centerline of the tube a distance y, along the tube, Fahrenheit.
$t_{s/2}$ = temperature of the facing sheet at a distance $s/2$ from the tube, Fahrenheit.

test results, however, revealed that no correction should be made for the loss in face area due to the perforations. Consequently, plain and perforated panels were treated alike in calculating the performance values shown in Table 1. Even then, the thermal performance of perforated panels exceeded slightly that of plain panels in four of the five cases in which direct comparisons can be drawn. It is believed that the cylindrical surface inside each perforated hole compensated for the loss in face area. Such a balance might not result for other combinations of face sheet thickness, hole size, and spacing.

Efficient transfer of heat through the brazed joint resulted in relatively small temperature differences between water in the tube and face sheet immediately under the tube. The average temperature difference for all heating and cooling tests, excluding those conducted on the somewhat impractical panel with the ½-in. thick face sheet, was 2.4 deg. This subject is discussed in more detail later.

THEORETICAL ANALYSIS

Direct application of test results is usually complicated by differences between test and service conditions. This is particularly true when performance is

NOMENCLATURE *(Concluded)*

t_N = arithmetic weighted average of all temperatures, t_t and $t_{s/2}$, observed on the longitudinal centerline of the panel, Fahrenheit.

t_n = arithmetic average of temperatures, t_t and $t_{s/2}$, at Section A-A, Fahrenheit.

t_M = mean temperature of the entire panel surface, Fahrenheit.

t_m = mean temperature of the panel surface at Section A-A, Fahrenheit.

c = heat transfer coefficient for the flow of heat from the water in the tube through the water film and metallic joint to the panel surface directly under the tube, Btu per (hour) (foot of tube) (Fahrenheit degree).

h = general heat transfer coefficient combining radiation, convection, and conduction, and relating heat flow to the difference between the first powers of t and t_r, Btu per (hour) (square foot) (Fahrenheit degree).

h_N = approximate heat transfer coefficient obtained from panel tests by relating average unit heat transfer rate, Q/A, to temperature difference between t_N and t_r, Btu per (hour) (square foot) (Fahrenheit degree).

h_M = adjusted heat transfer coefficient obtained from h_N by correcting for the difference between t_N and t_M, Btu per (hour) (square foot) (Fahrenheit degree). Represents experimentally determined h.

h' = modified heat transfer coefficient for Section A-A referred to temperature, t_t, directly under the tube, Btu per (hour) (square foot) (Fahrenheit degree).

h'' = modified heat transfer coefficient for Section A-A referred to water temperature, t_w, Btu per (hour) (square foot) (Fahrenheit degree).

h''' = overall heat transfer coefficient for the entire panel referred to the inlet water temperature, t_{w1}, Btu per (hour) (square foot) (Fahrenheit degree).

$$R = \frac{\tanh \sqrt{\frac{s^2 h}{4 k b}}}{\sqrt{\frac{s^2 h}{4 k b}}}$$

(tanh is the hyperbolic tangent).

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affected in varying degrees by as many variables as are associated with this problem. The following analytical procedure, therefore, has been developed for use in applying the preceding test results to a wider variety of ceiling panels and operating conditions than were included in the tests.

In this analysis, the nomenclature employed and the units of measure are as indicated unless otherwise noted.

Maximum heating and cooling performance of water-carrying ceiling panels having fixed surface characteristics would be attained if the surface temperature of the panel t were uniform throughout and equal to the temperature of the entering water, t_{w1}. This condition would result if it were possible to flow a sheet of heated or chilled water at infinite velocity over the top surface of a metal panel having no resistance to heat transfer through its thickness. Such maximum performance is partially sacrificed, in the tube-and-sheet type panels with which this paper is concerned, to gain other advantages. For example, conveying the water in tubing attached to the facing sheet provides ample space for acoustical perforations. Control of the water flow rate causes a change in thermal performance of the panels and can be applied to advantage due to the manner in which the performance requirements vary with respect to distance from outside building walls.

The tubing and facing sheet could be joined by brazing, welding, soldering, adhesive bonding, or mechanical connections. The temperature difference between the water and the facing sheet directly under the tube would be affected by the thermal resistance of such joints. Consequently, in this analysis, a coefficient c is used for the rate of flow of heat from the water in the tube through the water film and metallic joint to the panel surface directly under the tube.

The rate at which heat is released or absorbed by every unit area of a panel having a uniform surface temperature t can be expressed as,

$$Q/A = h\,(t - t_r) \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

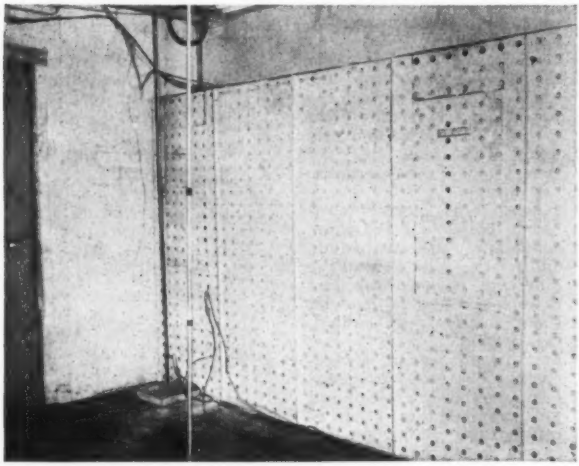


FIG. 4. INTERNAL VIEW OF ONE CORNER OF TEST ROOM SHOWING PERFORATED AIR DIFFUSING PARTITION

and the value selected for the heat transfer coefficient h depends upon considerations given to air motion, surface emissivity, insulation above the panel, and the general level of temperatures t and t_r.

For an aluminum tube and sheet panel in which tube spacing s, the facing sheet conductance bk, and water flow rate G have finite values, the face sheet temperature, t, varies with distance along as well as away from the tube. As was the case for the uniform temperature panel, the heat transfer coefficient h might be affected by the general level of the mean surface temperature, t_M, but it is assumed that a single value of h can be used for the entire panel. In order to calculate the thermal performance of such a non-uniform temperature panel, it is necessary to determine the temperature distribution on the panel surface.

Consider, therefore, an idealized type of heating panel, such as the one shown in the upper part of Fig. 5, in which the tube traces a square-cornered sinuous path. The marginal distance $s/2$ at all ends and edges of the panel is one-half the tube spacing s. The facing sheet is slit midway between adjacent parallel tube passes, as shown by the dashed lines. Straightening the sinuous tube to

TABLE 1—SUMMARY OF ALUMINUM HEATING AND COOLING PANEL PERFORMANCE†

			Q/A = PANEL THERMAL PERFORMANCE, BTU/(HR)(SQ FT)[7]															
			HEATING[3]												COOLING[4]			
NOMINAL INLET WATER TEMP			180° F				160° F				120° F				60° F			
NOMINAL AIR VELOCITY[2]			25 fpm		< 10 fpm		25 fpm		< 10 fpm		25 fpm		< 10 fpm		25 fpm		< 10 fpm	
NOMINAL WATER FLOW RATE			1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm	1 gpm	¼ gpm
TUBE SPACING	FACE SHEET THICKNESS	SURFACE EMISSIVITY																
8"	⅛"	94%	176 *182*		147 *142*	110 *107*	138 *147*		114 *115*	87 *87*			49 *61*	41 *46*	−26 *−29*		−25 *−28*	−18 *−19*
		(1) 94%	180 *182*	126 *128*			140 *147*	98 *104*			73 *78*	48 *55*			−30 *−29*	−20 *−20*		
		91%	180 *181*	121 *128*	140 *140*	106 *106*	146 *146*	101 *104*	104 *113*	82 *86*	77 *77*	52 *55*	50 *60*	38 *45*	−25 *−29*	−20 *−20*	−23 *−28*	−18 *−19*
		69%	167 *169*	123 *122*	127 *122*	98 *97*	131 *137*	97 *99*	109 *99*	82 *78*	74 *72*	50 *52*	59 *52*	39 *41*	−20 *−27*	−19 *−19*	−22 *−26*	−17 *−18*
		(1) 33%	138 *149*		103 *95*		128 *121*		75 *77*	65 *64*			36 *41*		−20 *−24*		−26 *−22*	
20"	⅛"	94%			120 *107*	97 *85*	121 *107*		97 *87*	76 *69*			51 *46*	38 *36*	−20 *−21*		−21 *−20*	−16 *−14*
		(1) 94%	135 *132*	108 *98*			111 *107*	82 *79*			59 *57*	44 *42*			−23 *−21*	−18 *−14*		
		91%	132 *131*	100 *98*	119 *107*	93 *84*	107 *106*	84 *79*	94 *87*	70 *68*	51 *56*	43 *42*	45 *46*	36 *36*	−23 *−21*	−18 *−14*	−24[6] *−25*[6]	−18[5] *−16*[5]
		69%	130 *125*	105 *93*			110 *101*	86 *76*			56 *54*	43 *40*			−22 *−20*	−17 *−14*		
20"	3/32"	33%							66 *60*	56 *50*								
		4%			61 *63*	57 *56*			53 *51*	42 *45*			30 *27*	22 *24*			−14[6] *−17*[6]	−13[5] *−12*[5]
20"	½"	94%			134 *133*	103 *99*			119 *108*	81 *80*			61 *57*	41 *42*			−33 *−29*[5]	−19 *−19*[5]

† All panels 4 ft 3 in. × 14 ft 0 in. except 3/32 in. thick panels which were 4 ft 3 in. × 13 ft 6 in.; all insulated with 1 in. thick glass wool blanket covered with aluminum foil on the top side

(1) These panels perforated over entire surface with 1/16 in. diameter holes on ½ in. centers (perforations occupy 5 percent of total panel area)

(2) Near face of panel

(3) t_a = 70 F, t_r = 75 F

(4) t_a = 80 F, t_r = 77 F (except as noted)

(5) t_r = 79 F

(6) t_r = 81 F

t_a = Average controlled room air temperature

t_r = Mean wall and floor temperature

(7) Test results shown—176, 147, etc.

Analytical results shown—182, 142, etc.

its developed length D results in the single straight tube panel shown in the upper part of Fig. 6. Inasmuch as this represents one element of a parallel grid system, the analysis presented here applies to either parallel or series flow arrangement of the water-carrying tubes.

Section A-A, as shown in the lower part of Fig. 5, represents a typical cross section of the idealized or straight tube panel. In effect, the facing sheet now becomes two symmetrical fins attached to the tube. Mathematical consideration

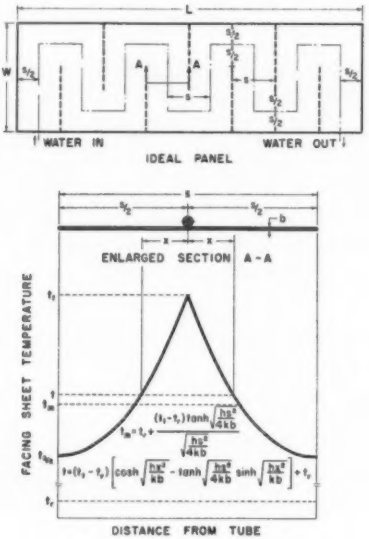


FIG. 5. IDEALIZED PANEL WITH TEMPERATURE DISTRIBUTION AT TYPICAL CROSS SECTION

of the heat flow through and from each fin results in the temperature distribution indicated by the curves shown under the enlarged Section A-A in Fig. 5 and expressed by the equation,

$$t = (t_t - t_r)\left[\cosh\sqrt{\frac{h\,x^2}{k\,b}} - \tanh\sqrt{\frac{h\,s^2}{4\,k\,b}}\,\sinh\sqrt{\frac{h\,x^2}{k\,b}}\right] + t_r \quad . \quad . \quad . \quad (2)$$

in which cosh, tanh, and sinh designate hyperbolic functions. Similarly, the mean surface temperature, t_m, for Section A-A can be expressed in the equation,

$$\frac{(t_m - t_r)}{(t_t - t_r)} = \frac{\tanh\sqrt{\frac{s^2\,h}{4\,k\,b}}}{\sqrt{\frac{s^2\,h}{4\,k\,b}}} = R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The rate at which total heat per unit area is released from the panel at Section A-A is,

$$q_{A\text{-}A} = h\,(t_m - t_r) \quad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

which can be expressed,

$$q_{A\text{-}A} = hR\,(t_t - t_r) \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

During heating service, the temperature of the water, t_w, decreases as the water flows through the tube and this results in a similar decrease in panel surface temperatures, t_t and t_m, as indicated by the curves in the lower part of Fig. 6. Consequently, the rate of total heat output per unit area, $q_{A\text{-}A}$, also decreases from the inlet to outlet end of the panel, and it is necessary to integrate

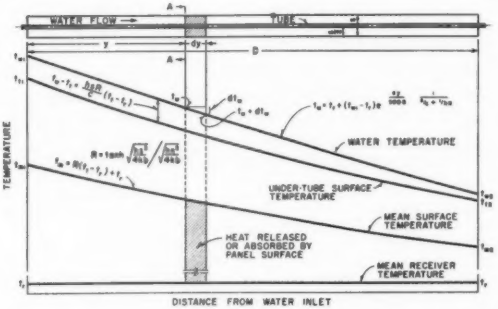


FIG. 6. DEVELOPED SINGLE TUBE PANEL WITH TEMPERATURE DISTRIBUTION CURVES

this effect along the tube in order to determine the thermal performance of the entire panel.

The water, in passing through the incremental length of panel, dy, shown in Fig. 6, loses an amount of heat equal to that released by this short length of panel. This condition is expressed by the differential equation,

$$-\,w\,p\,\frac{d\,t_w}{dy}\,dy = hs\,(t_m - t_r)\,dy \quad \ldots\ldots\ldots\ldots \quad (6)$$

and in terms of the temperature drop $(t_w\text{-}t_t)$, across the joint,

$$-\,w\,p\,\frac{d\,t_w}{dy}\,dy = c\,(t_w - t_t)\,dy \quad \ldots\ldots\ldots\ldots \quad (6a)$$

where c is the conductance of the joint per unit length of tubing.
Because of the fact that:

1. $wp = 500G$
2. $hs\ (t_m - t_r) = c\ (t_w - t_t)$, by combining Equations 6 and 6*a*
3. $(t_m - t_r) = R\ (t_t - t_r)$, from Equation 3

Equation 6 can be converted to the form,

$$-\,500\,G\,\frac{d\,t_w}{dy} = \frac{hs\,Rc}{hs\,R + c}\,(t_w - t_r) \quad \ldots\ldots\ldots \quad (6b)$$

Solution of this differential equation results in the following expression for t_w at any distance y measured along the tube from the inlet end where $t_w = t_{w1}$

$$t_w = t_r + (t_{w1} - t_r)\, e^{-\frac{s\,y}{500\,G} \cdot \frac{h\,R\,c}{h\,R\,s + c}} \quad \ldots\ldots\ldots \quad (7)$$

From this it follows that, at the outlet end of the panel,

$$t_{w2} = t_r + (t_{w1} - t_r)\, e^{-\frac{s\,D}{500\,G} \cdot \frac{1}{s/c + 1/h\,R}} \quad \ldots\ldots\ldots \quad (8)$$

Under equilibrium conditions, heat is released from the entire panel at the same rate as it is lost by the water; consequently,

$$Q = 500G\,(t_{w1} - t_{w2}) \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

Combining Equations 8 and 9,

$$Q = 500\,G\,(t_{w1} - t_r)\left(1 - e^{-\frac{s\,D}{500\,G} \cdot \frac{1}{s/c + 1/h\,R}}\right) \quad \ldots\ldots \quad (10)$$

The average rate of heat output per unit area for the straight tube panel becomes,

$$\frac{Q}{A} = \frac{500\,G}{s\,D}\,(t_{w1} - t_r)\left(1 - e^{-\frac{s\,D}{500\,G} \cdot \frac{1}{s/c + 1/h\,R}}\right) \quad \ldots\ldots \quad (11)$$

and for the sinuous tube panel,

$$\frac{Q}{A} = \frac{500\,G}{W\,L}\,(t_{w1} - t_r)\left(1 - e^{-\frac{W\,L}{500\,G} \cdot \frac{1}{s/c + 1/h\,R}}\right) \quad \ldots\ldots \quad (12)$$

in which R is a function of h, s, k, and b, as indicated in Equation 3.

The rate of heat transfer is most commonly expressed by a coefficient times a temperature difference. Thus, for a uniform temperature panel for which the rate of heat transfer is expressed by Equation 1, the coefficient is h and the temperature difference is $(t\text{-}t_r)$.

The rate of heat transfer per unit area at any Section A-A of a tube and sheet panel is expressed by Equation 5 in which the coefficient, which will be denoted by h', is hR and the temperature difference is $(t_t\text{-}t_r)$. Note that in this case, t_t is the temperature at only one point in the section and that the reduction factor R is imposed upon the general coefficient h to compensate for the fact that for panel heating, all other points in the section are at temperatures lower than t_t. The coefficient, h', which applies to the temperature difference $(t_t\text{-}t_r)$ can, therefore, be stated as,

$$h' = hR = h\,\frac{tanh\sqrt{\frac{s^2\,h}{4\,k\,b}}}{\sqrt{\frac{s^2\,h}{4\,k\,b}}} \quad \ldots\ldots\ldots \quad (13)$$

and Equation 5 becomes,

$$q_{A\text{-}A} = h' (t_t - t_r) \quad \ldots \ldots \ldots \ldots \quad (14)$$

The heat transfer rate per unit area at any Section A-A can also be expressed in terms of the temperature difference $(t_w\text{-}t_r)$, by defining the coefficient, h'', which is actually the sum of the thermal coefficient of conductance of the joint plus the coefficient, h'. So,

$$h'' = \frac{1}{s/c + 1/h'} \quad \ldots \ldots \ldots \ldots \quad (15)$$

and

$$q_{A\text{-}A} = h'' (t_w - t_r) \quad \ldots \ldots \ldots \ldots \quad (16)$$

For the entire tube and sheet panel, the average rate of heat transfer per unit area can be expressed as,

$$Q/A = h''' (t_{w1} - t_r) \quad \ldots \ldots \ldots \ldots \quad (17)$$

In this case, the temperature t_{w1}, is higher for panel heating than the temperature at any other point in the panel. Consequently h''' must result from a triple reduction imposed on h; one for the temperature drop across the joint; one for the temperature drop which occurs as the point moves away from the tubes; the other associated with the point moving along the tube path. The coefficient, h''', obtained from Equation 12 is,

$$h''' = \frac{500\,G}{W\,L}\left(1 - e^{-\frac{W\,L}{500\,G}\cdot\frac{1}{s/c + 1/h\,R}}\right) \quad \ldots \ldots \quad (18)$$

From this it can be seen that the first reduction operates through the relationship between the tube spacing s and the joint conductivity c, the second reduction operates through R, and the third reduction is a function of the relationship between water flow rate G and the area of the panel WL.

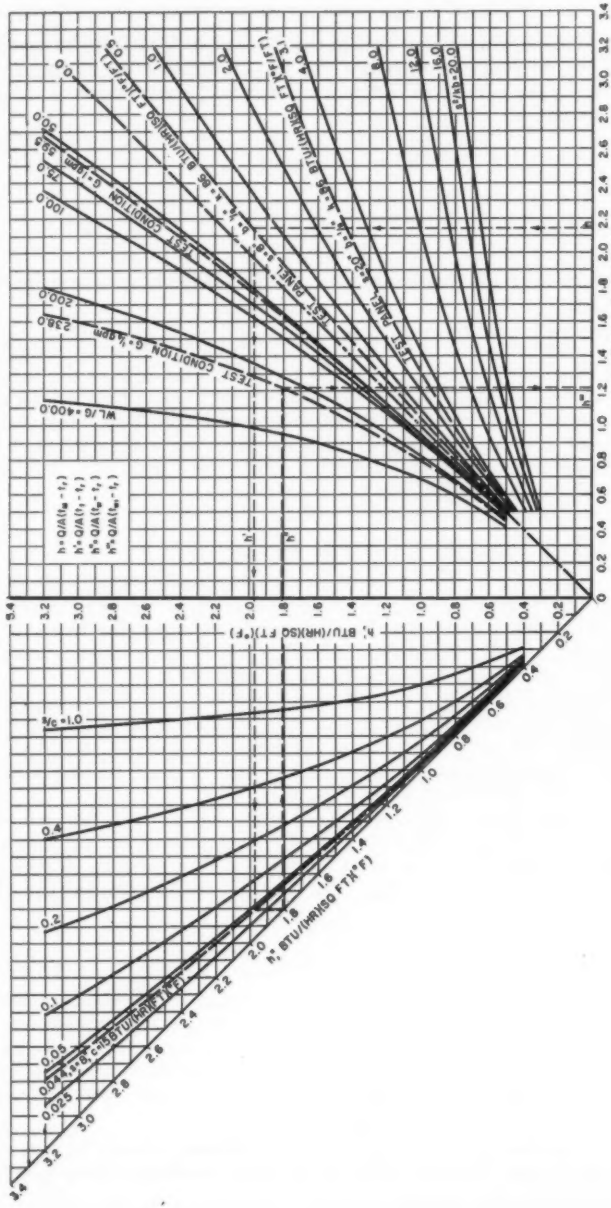
Use of Equations 17 and 18 permits calculation of the average rate of total heat output per unit area for a panel through the substitution of values for all pertinent design and operating factors. Whereas the analysis was developed on the basis of panel heating, it applies equally well to panel cooling, in which case the heat flow would be a negative quantity.

In order to apply Equations 17 and 18, proper values must be determined for c in terms of flow rate and temperature and for h in terms of surface emissivity, air flow, and insulation above the panel. Analytical determination of these factors is not within the scope of this paper, but the values obtained from tests are presented later.

DESIGN CHART

Calculation of h''' by using Equations 13, 15, and 18 is somewhat tedious and it is difficult to sense the relationship between h''' and h as determined by the other variables. Unfortunately, the reduction factors previously discussed are, in themselves, functions of h as well as the other variables, but fortunately the reducing operation can be accomplished readily in three steps. The design chart presented in Fig. 7 can be used for this purpose.

Example 1. Calculate the overall heat transfer coefficient referred to inlet water temperature for a panel operating under the following conditions: $s = 8$ in. $= \frac{2}{3}$ ft;


TEST CONDITION G = 1/4 GPM
TEST PANEL = 22"x5'-0" x 1/16" ALUMINUM SHEET
TEST CONDITION G = 1/2 GPM
TEST CONDITION G = 1 GPM
W L/G = 400.0
200.0
100.0
50.0
25.0
12.5
6.25
3.12
1.56
0.78
0.39
0.20
0.10
h = Q/A(t_w - t_r)
h' = Q/A(t_i - t_r)
h'' = Q/A(t_w - t_r)
h''' = Q/A(t_w - t_r)
h, BTU/(HR)(SQ FT)(°F)
h' BTU/(HR)(SQ FT)(°F)
h'' BTU/(HR)(SQ FT)(°F)
h & h'', BTU/(HR)(SQ FT)(°F)
h'''
h'
h''
0.1" 3/4"
0.4
0.2
0.1
0.05
0.04
0.025
0.02
0.01
0.008
0.006
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$k = 86$ Btu per (hr) (sq ft) (F deg per ft); $b = \frac{1}{8}$ in. $= \frac{1}{96}$ ft; $c = 15$ Btu per (hr) (ft of tube) (F deg); $W = 4.25$ ft; $L = 14.00$ ft; $G = \frac{1}{4}$ gpm.

Solution. In order to use the design chart, Fig. 7, it is first necessary to calculate s^2/kb, s/c and WL/G which will be 0.5, 0.044 and 238, respectively.

If it is assumed that h is known, h''' can be obtained from the design chart by following the course indicated by the dashed lines. The chart is entered from the right hand abscissa at the value of h, such as 2.15 for the example. The vertical dashed line is followed until it intersects the curve for which $s^2/kb = 0.5$, at which location it can be observed that $h' = 1.97$ on the ordinate. This ordinate line is followed to the left until it intersects the curve for $s/c = 0.044$. By dropping vertically to the inclined base line h'' is found to be 1.81. Proceeding horizontally to the right to intersect the curve for $WL/G = 238$ and dropping vertically, the value of $h''' = 1.22$ is found on the abscissa. This value for h''' is approximately 57 percent of that assumed for h.

Example 1 was selected as representing one of the test panels and will be referred to later when the heat output of the panel is calculated.

EXPERIMENTAL DETERMINATION OF HEAT TRANSFER COEFFICIENTS, h AND c

The theoretical analysis can now be checked with the experimental results, but the experimental data must first be reduced to the terms of the theory.

The choice of an experimental value for the mean heat receiver temperature, t_r, is a difficult one, since t_r should actually be a weighted mean of several air and surface temperatures. Heat is lost from the panel to the room air by convection, to the walls and floor by radiation, and to the space above the ceiling by conduction, convection, and radiation. The proportions lost in each manner are influenced by panel emissivity, air velocity, and probably panel temperature. However, in this work, the experimental value of mean receiver temperature, t_r, was assumed to be equal to the weighted mean of ten measured temperatures on the walls and floor of the test room. An average panel surface temperature, t_N, was obtained by taking a weighted average of six measured surface temperatures at points directly under the tube and four points midway between tubes.

With values for Q/A, t_r, and t_N, it was possible to obtain an experimental value for the heat transfer coefficient, h_N, from Equation 1, employing the calculated value of t_N in place of t. However, it can be seen from the theoretical analysis of a typical cross section that the value of average surface temperature, t_n, may be considerably different from the actual mean surface temperature t_m because the temperature distribution between tubes follows a logarithmic path rather than the straight line assumed in determining t_n. Therefore, a relationship between the actual mean surface temperature t_m and the measured average temperature t_n was determined from Equation 2, and a similar relationship was assumed to exist between temperature t_M and t_N for the whole panel. Thus:

$$\frac{t_m - t_r}{t_n - t_r} = \frac{2 \sinh \sqrt{\frac{s^2 h}{4kb}}}{\sqrt{\frac{s^2 h}{4kb}}\left[1 + \cosh \sqrt{\frac{s^2 h}{4kb}}\right]} = \frac{t_M - t_r}{t_N - t_r} = \frac{h_N}{h_M} \quad . . (19)$$

By this relationship, h_N was converted to h_M (which approximates h of the theoretical analysis) for each test.

The experimental values of h_M determined from Equation 19 represent several panel surface emissivities and two air flow conditions. In order to show the effect of emissivity and air flow velocity on the heat transfer coefficient, Fig. 8 was constructed by plotting average values of h_M against emissivity for each of the two air flow conditions. The data points in Fig. 8 represent the average of values of h_M for all tests run at each condition of emissivity and air flow.

The lower of the two solid curves in Fig. 8 represents data from tests made with the perforated air diffuser partition erected along the inlet air wall of the test room. With this arrangement, the air flow velocity was considered to be less than 10 fpm. The upper solid curve in Fig. 8 represents data from tests made with no diffuser partition and with the louvers on the inlet air grilles

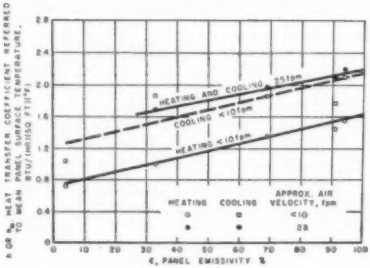

FIG. 8. VARIATION OF EXPERIMENTAL HEAT TRANSFER COEFFICIENT WITH SURFACE EMISSIVITY AND AIR VELOCITY

turned half up and half down. With this arrangement, the average of air velocities was 25 fpm.

The dashed line in Fig. 8 represents the variation of h_M with emissivity, ϵ, for cooling tests involving inlet water temperatures of 60 F and air flow velocities of less than 10 fpm. That the heat transfer coefficient is higher for cooling than heating is explained by the fact that in cooling the natural convection causes the warm air to continually rise and replace the cooler air next to the cold panel. In the case of heating panels, there is a layer of very warm air against the ceiling panel which tends to remain near the ceiling and this lack of air motion retards the release of heat by convection.

This conclusion is supported by the fact that the coefficients for heating and cooling are nearly identical for the tests at the higher air velocity of 25 fpm, Fig. 8. In this case, there is sufficient forced air movement to cause removal of the layer of warm air at the ceiling, thus increasing the convection heat transfer. Thus, the heat transfer coefficient is nearly identical for heating and cooling when there is appreciable air motion across the panel surface.

A thorough study of the effect of emissivity, air motion, and temperature upon the surface heat transfer coefficient h was beyond the scope of the work reported in this paper. However, the values shown in Fig. 8 are believed to be typical of those to be expected in ceiling panel installations.

The coefficient of heat transfer from the water in the tube to the panel surface

under the tube c was also determined from the experimental data. This coefficient was calculated by the equation:

$$c = \frac{(Q/A)\ (s)}{\Delta t} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (20)$$

The heat transferred per unit area of panel, Q/A, was determined for each test, as previously described. The average temperature difference, Δt, was determined for the whole panel by subtracting the average of six measured under-tube surface temperatures from the average of three measured water temperatures for each test.

Considerable variation occurred among the values calculated for this coefficient. However, Equation 20 shows that for a given heat transfer rate, the value of c varies inversely with the temperature difference. With this type of relationship, for small values of Δt, such as were generally encountered in these tests, a small change in Δt will produce a considerable change in c. In these tests, the measured values of Δt ranged from zero to approximately 10 deg with an average of 2.4 deg for all tests. Thus, small errors in the measurements of Δt could be expected to produce rather wide variations in the values obtained for c. Actual values of c varied from approximately 4 to 120 Btu per (hr) (ft of tube) (F deg) with the average level of c being influenced by flow rate and temperature.

Although there was considerable variation in the experimentally determined values of c, it can be shown by the analytical equation that, for a large change in c above about 10 Btu per (hr) (ft) (F deg) there is a very small change in the heat transfer rate, h'''. Therefore, the following average values of c, Btu per (hr) (ft) (F deg) were determined for heating and cooling at the indicated water flow rates:

	1 GPM	¼ GPM
Heating	50.0	15.0
Cooling	25.0	7.5

These values of c are believed to be sufficiently accurate for a brazed joint that provides a short, wide heat path of high conductivity. However, with other types of joints that provide a lower overall conductivity, more accurate determinations of c would probably be necessary, since the overall heat transfer rate, h''', becomes quite sensitive to variations in c at low values of c.

COMPARISON OF EXPERIMENTAL AND CALCULATED PERFORMANCE

Theoretical panel performance Q/A for each of the test conditions in Table 1, was determined through the use of the design chart, Fig. 7, in accordance with the previously described method using values of h selected from Fig. 8 and the values of c reported in the previous section.

For example, the panel used to demonstrate the application of the design chart had been painted with an ivory paint that gave a surface emissivity of 94 percent. With the test room inlet air duct louvers adjusted half up and half down, the average air velocity across the panel surface was 25 fpm. For this emis-

sivity and air velocity condition, the heat transfer coefficient h from Fig. 8, was found to be 2.15, the value used in the design chart example. When the panel was used for heating at a water flow rate of ¼ gpm, the joint heat transfer coefficient c was shown in the previous section to be 15.0. Using these values of h and c, the overall heat transfer coefficient h''' was found from the design chart to be 1.22. Q/A was then determined by multiplying h''' by $(t_{w1}\text{-}t_r)$. The average value of t_r for all heating tests was 75 F. Consequently, for inlet water temperatures of 180, 160, and 120 F, the values of $(t_{w1}\text{-}t_r)$ were 105, 85, and 45 deg, respectively. Multiplying these values by $h''' = 1.22$ gave values for Q/A of 128, 104, and 55 Btu per (hr) (sq ft). Similar calculations were made for all test conditions and are presented in Table 1 as the lower value in each block.

In general, the agreement between actual test results and calculated performance improved with increasing panel temperature. As might be expected, the greatest differences occurred for the cooling tests. On the whole, 46 percent of the calculated values were within ±5 percent of the measured values, 75 percent were within ±10 percent, and 90 percent were within ±15 percent. In the worst case, the theoretical value was 33 percent greater than the experimental.

The fact that the actual values of the heat transfer coefficient varied considerably from the average values used in the calculations is the primary cause of differences between test and calculated performance. When the actual experimental value of h_M for each test was used to calculate performance, 60 percent of the calculated values were within ±5 percent of the measured values, 93 percent were within ±10 percent, and the greatest deviation was 16.7 percent. Substitution of actual values of c and $(t_{w1} - t_r)$ in the calculations further improved the agreement.

CONCLUSION

It can be concluded that reasonably accurate predictions of the thermal performance of a wide variety of aluminum ceiling-type heating and cooling panels, (including types in which aluminum panels are fastened to tubes by mechanical means such as clips) can be made through the use of the design chart (Fig. 7) or Equations 11 or 12 and properly selected values of all contributing factors.

ACKNOWLEDGMENT

The authors express their appreciation for assistance in outlining the test program to the architects for the Alcoa Pittsburgh Office Building, Harrison and Abramovitz, and to the consulting engineers, Jaros, Baum and Bolles. Alfred L. Jaros, Jr. was particularly helpful throughout the work. Design and construction of the panels and test room facilities were supervised by Aluminum Company of America staff engineers, E. C. Collins, P. J. McCarthy, E. V. Cross, B. R. Small, H. M. Short, and L. M. Young. Determination of values of thermal conductivity and emissivity were made under the direction of C. S. Taylor, chief, Physical Chemistry Division, Aluminum Research Laboratories.

DISCUSSION

C. S. LEOPOLD, Philadelphia, Pa. (WRITTEN): The data on panel cooling in this paper apply to the specific conditions of the tests; namely, a given temperature

difference between room air and panel, a given temperature difference between mean radiant temperature of the room and panel, and a load consisting in part of recessed fluorescent lights and in part of warm air.

If the load were changed to consist of surface mounted partly indirect fluorescent lights, or of filament lighting only, the required water temperature to accomplish the same heat removal would be considerably closer to the room air temperature than indicated in this paper. If a portion of the heat removal were accomplished by the cold air which may be used for ventilation, there would be a more than proportionate decrease in the required difference in temperature between panel and room air which would have the effect of producing a large increase in the apparent value of h. The value of h in this paper, for an emissivity of 0.9, is given as 2.1. In our tests, as reported to *A.S.R.E.* in 1947 [2], for a completely cooled ceiling with no heat released above the panels, the apparent value of h varied from 1.89 to 14.3, depending on the type of luminaire used, the percentage of the work which was done by the ventilating air, and the surface finish in regard to its absorption for the radiation from a luminaire.

In a paper presented to *A.S.R.E.* in 1948 [3], we reported our findings on the effect of the solar load, using a scale model of a building, an artificial sun, and a completely cooled ceiling. In this case the apparent value of h varied from a low of 2.18 to a high of 2.88, depending upon the type of glass and shading device used. These values are significantly higher than those published in this paper and lead to the conclusion that, if the actual nature of the load is analyzed, either less panel or panels at higher temperature can usually be used.

In a paper presented to THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in January, 1951[4], we presented the data on ceilings which were cooled in part. It was considered that in the over-all picture, some use should be made of the top surface of the panel. Acoustical engineers have stated that for office use ample sound correction could be obtained with only 40 percent of the ceiling in the form of perforated panels. By exposing the top of the panel, the panel can remove thermal energy from the slab of the floor above, a portion of the energy which is transmitted through the uncooled portions of the ceiling and energy released from recessed luminaires. For the part paneled ceiling and a load due to solar radiation the tests indicated apparent values of h varying from 3.26 to 4.22.

Tests in the original cubicle for a load due to artificial illumination with 29 percent of the ceiling in the form of cooling panels, show an apparent h varying from 2.54 to 4.29, depending on the type of luminaire and the chemistry of the white paint.

A cooling panel which in part formed the reflector for a troffer fluorescent light is also shown, and here the direct interception of radiation, conduction and restrained convection results in a still higher fictitious value of h.

It appears desirable to emphasize that the performance of a cooling panel depends not only on its emissivity at low temperature but on its absorptivity for the radiation emanating from the heat load and that, in general, the larger the portion of the heat load which is initially in the form of radiation, the higher the apparent performance of the panel. The concept of $h = h_r + h_c$ as a constant which may be used over a wide range of conditions is not correct unless a separate analysis is made of the relatively high temperature radiation from the heat causing source which actually reaches the cooled panel, and the mean radiant temperature of the surfaces *seen* by the panels is either measured or calculated.

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[2] Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by C. S. Leopold (*Refrigerating Engineering*, July 1947).

[3] Mechanism of Heat Transfer, Panel Cooling, Heat Storage; Part II, Solar Radiation, by C. S. Leopold (*Refrigerating Engineering*, June 1948).

[4] Design Factors in Panel and Air Cooling Systems, by C. S. Leopold (see Chapter 1409, p. 61).

I believe an over-simplification of this problem will lead to the selection of more cooling surfaces or greater temperature differences between room air and panels than are actually required.

J. E. MURRAY, San Francisco, Calif.: Would the finished appearance of a panel having an emissivity of 0.94 be acceptable esthetically?

L. E. SEELEY, Durham, N. H.: You indicated two methods of introducing air, but did not refer to both in your slides.

AUTHORS' CLOSURE (E. S. Howarth): With reference to Mr. Murray's question, the ceiling panels having an emissivity of 94 percent do present a good appearance. This value was obtained through the application of three finish coats of interior flat ivory paint. The light appearance was similar to that observed for most ceilings, even those which are not treated for thermal or acoustical performance. It is not necessary for the ceiling to be dark in color when its thermal performance is desired to approach that of a "black body".

The two different methods of introducing air, mentioned by Dean Seeley, were utilized to evaluate the effect of air motion on the convective component of the total heat released or absorbed by the ceiling panels. Reference to Fig. 8 indicates that the values for h are functions of air flow as well as emissivity and that the effect of variations in air flow was greater for heating operation than for cooling.

Mr. Leopold's discussion and caution concerning the selection of proper values for h are in accord with the authors' thinking on this subject. He is correct in saying that the data presented in this paper apply to the specific conditions of these particular tests. In fact, this investigation was initiated because of a fear that available values of h might be too high to be used for conditions which would exist in the 30-story office building under consideration. In this building, the solar load is diminished through the use of relatively small window areas and by the heat absorbing glass which is utilized in the double glazed window construction. It was also known that recessed fluorescent lighting fixtures would be installed flush with the ceiling.

Determination of h involves consideration of the mean temperature of the heat source, t_r. In this work, t_r was the weighted mean temperature of those floor and wall surfaces *seen* by the ceiling panels. The average air temperature was observed to have approximately the same value as t_r. However, in additional tests, not reported in this paper, auxiliary side wall panels were utilized to provide local heat sources. When their temperatures and areas were incorporated in the weighted value for t_r, the values determined for h did not change appreciably. In Mr. Leopold's work, described in his Reference (2), apparent values of h varied from 1.89 to 14.3, but the value used for t_r was the measured air temperature at the 5-foot level. This method gave no weight to the radiating temperatures of the floor and walls, and excluded the important contribution of the position, area, and temperature of the luminaire. These factors contribute to the wide variation in apparent values for h. Those values of h reported by Mr. Leopold for conditions in which both the top and bottom surfaces of the ceiling panels were active are, as might be expected, about twice as great as those reported in this paper for panels which were insulated on the top side.

The authors do not minimize the difficulties encountered in determining true values for h and believe that use of apparent values simplifies the analytical procedure, provided that such values are available for the design conditions.

1426

No. 1426

LABORATORY STUDIES OF THE THERMAL CHARACTERISTICS OF PLASTER PANELS

By C. M. HUMPHREYS*, C. V. FRANKS**, AND L. F. SCHUTRUM***,
CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THE SOCIETY is engaged in a comprehensive research program on panel heating and cooling. The work described in this report was conducted under the general direction of Group A, Technical Advisory Committee on Panel Heating and Cooling. This group has the responsibility for investigations of heat transfer and temperature distribution within, and behind, the panel.

It is recognized that the heating panel is only one element of the complete thermal circuit in a panel heating system, and that the panel performance is influenced by the space and the five surfaces of the room which it *sees*. In keeping with the scope of the directing committee, this paper deals only with the heat flow to the panel face and the heat flow to the back of the panel. The modifying effects of the surroundings will be studied in a calorimeter room now being constructed at the Research Laboratory, under the direction of Group B of the Technical Advisory Committee.

This report presents the results of laboratory studies on four plaster panels, representing three conventional types of ceiling panels. Secondary effects, such as temperature variations on the panel surface, also are included.

PANEL CONSTRUCTION

The plaster panels were constructed on wood frames, 6 ft square, with joists in the test section located 16 in. on centers. The method of framing may be observed in Fig. 1, a view of the back of panel No. 3.

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Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

Each panel contained 17 tubes, or pipes, located 4 in. center to center. Details of panel constructions are given in Table 1 and Fig. 2.

Plastering was done by an experienced craftsman on the panel frames supported overhead in a horizontal position. Three-coats of plaster conforming to the standard specifications for gypsum plastering[1] were applied to each panel. The first (scratch) coat consisted of 1 part unfibered neat gypsum and 2 parts sand, by weight. The second (brown) coat consisted of one part

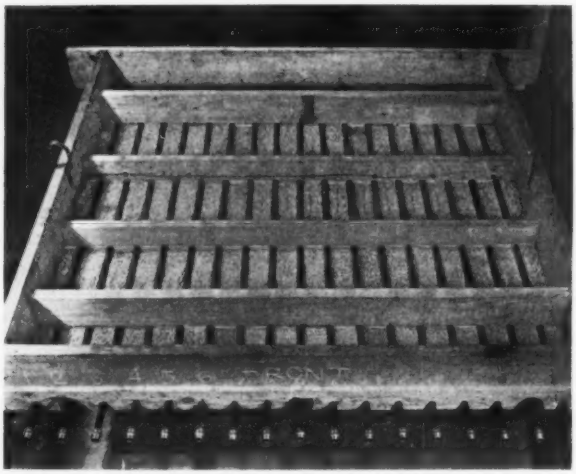

FIG. 1. BACK VIEW OF PANEL NO. 3 SHOWING FRAMING DETAILS

of unfibered neat gypsum and three parts of sand, by weight. The finish coat consisted of 3 parts lime putty and 1 part gaging plaster, by volume.

A *Gypsum Association* representative, who was familiar with field practice, was present to observe and give advice when the first panel was plastered, since it was important to obtain tube embedment typical of that found in the field.

TEST APPARATUS AND PROCEDURE

Test apparatus and procedures were essentially the same as those used for the concrete panel tests reported in a recent paper.[2] The panel to be tested was placed between two heat absorbers, as shown in Fig. 3. By maintaining the heat absorber temperatures at the proper levels, it was possible to control the heat flow rate from each side of the panel. The heat flow rates were measured by heat flow meters incorporated in the absorber construction. Guards were placed along the four sides of each absorber to eliminate edge losses.

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[1] Exponent numerals refer to References.

The tubes or pipes embedded in each panel were connected to headers to form a grid-type coil. A ground key cock was provided on each tube to permit, by selective valving, tests with 4, 8, 12, and 16 in. tube spacings. The temperature of the water to the panel could be held constant at any desired value.

A more comprehensive description of the heat absorbers, fluid circuits, and methods of measurement, is given in a recent paper[2] on concrete panels.

Panel temperatures were obtained by copper-constantan thermocouples located in a line running across the center of each panel at 90 deg to the tubes. For a width of eight tube spaces, the couples were located 1 in. apart as shown in Fig. 2. Across the remainder of the panel, couples on the face of the panel and on the lath were spaced 4 in. apart, each couple being in a vertical plane passing through a tube axis. Thermocouples on the finished panel surface were cemented

TABLE 1—DETAILS OF PANEL CONSTRUCTION

PANEL NO.	SIZE AND TYPE TUBING[a], IN.	LOCATION OF TUBES	TYPE OF LATH	DEPTH OF COVER[d] IN.
1	3/8 non-ferrous	Above lath	Expanded metal[b]	3/4
2	3/8 non-ferrous	Below lath	Expanded metal[b]	3/8
3	1/2 ferrous pipe	Above lath	Expanded metal[c]	3/4
4	3/8 non-ferrous	Below lath	Gypsum lath	3/8

a Non-ferrous tubing sizes are nominal. Ferrous pipe size is IPS.
b Flat rib type weighing 2.5 lb per square yard.
c Diamond key type weighing 3.1 lbs/ sq yd.
d Nominal thickness of plaster under tubes.

into small scratches in the plaster. Those on the tubes and metal lath were soldered in place.

In general, each series of tests was made with tube temperatures that were held constant, and with absorber temperatures that were varied to produce heat flow rates of the desired magnitude. Temperature distribution and heat flow rates from both sides of the panel were determined for each test.

Tests were made without insulation on back of panel, and also with 1½ in. or 3⅝ in. bat-type insulation laid on the back of the plaster. The nominal conductivity of the insulation was 0.27 Btu per (hr) (sq ft) (F deg per in.) Other tests were made with a double yellow pine floor on top of the joists approximately 6 in. above the back of the plaster.

After completion of the initial work on panel No. 1, back-plaster was applied to a depth of approximately 3/16 in. over the bare tubes, from which it was tapered down to the metal lath. The area covered by this back-plaster was approximately 60 percent of the area of the panel. A few tests were then made to compare panel performance with and without back-plaster.

METHOD OF ANALYSIS

Test data were correlated by developing an empirical relationship between panel output and the difference between the average tube temperature and the

average panel surface temperature. The relationship is expressed by the equation

$$\Delta T = aq_d + bq_u \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

where

ΔT = the difference between the average tube temperature and the average panel surface temperature, Fahrenheit degrees

q_d = rate of downward heat flow from the ceiling surface, Btu per (hour) (square foot)

q_u = rate of upward heat flow from the back surface, Btu per (hour) (square foot)

a and b = coefficients, the values of which depend upon tube size and spacing, type of lath, location of tubes with respect to the lath, degree of tube embedment, and other variables in the panel construction.

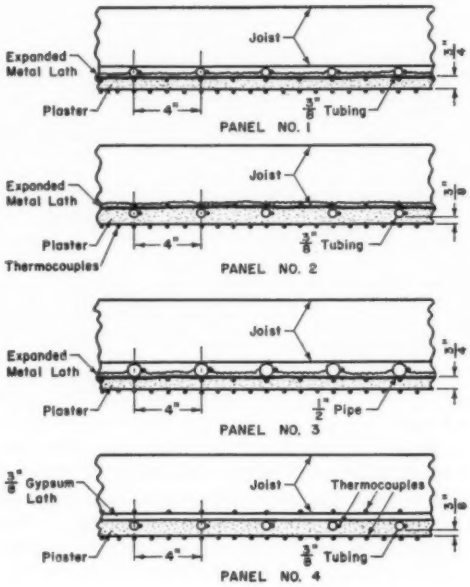

FIG. 2. PLASTER PANEL SECTIONS

ΔT, q_d, and q_u were determined experimentally in each test. For each pair of tests on a given panel with given tube spacing two equations were written. Each involved the same two coefficients, a and b. The values of a and b were determined by solving these pairs of equations simultaneously.

The coefficients a and b of Equation 1 were found to be almost independent of the amount of insulation for a given panel construction and tube spacing. It seemed feasible, therefore, to develop average values for these coefficients that

would be applicable for any condition of back insulation. These average values of a and b for the four panels tested are given in Figs. 4 and 5. The experimentally-determined temperature difference for 90 percent of the tests was within 15 percent of the values calculated from the curve values of a and b. While

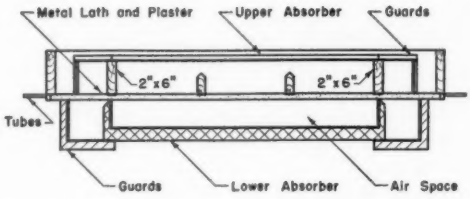

FIG. 3. SECTIONAL VIEW OF PLASTER PANEL AND ABSORBERS

somewhat closer agreement could have been obtained with a different set of curves for each insulation condition, the slight increase in precision that would have resulted did not seem to justify such procedure. It should be noted in Fig. 4 that the values of coefficients a and b for panels 1 and 3 are almost the

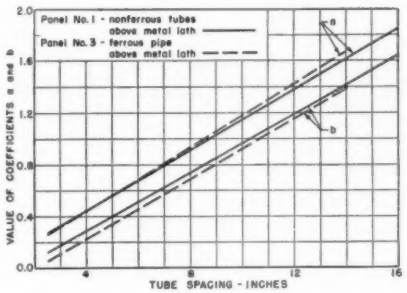

FIG. 4. COEFFICIENTS a AND b FOR EQUATION $\Delta T = aq_d + bq_u$: PANEL NOS. 1 AND 3

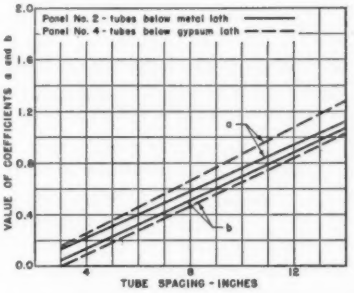

FIG. 5. COEFFICIENTS a AND b FOR EQUATION $\Delta T = aq_d + bq_u$: PANEL NOS. 2 AND 4

same. This is to be expected, since the panels are of the same general construction and differ only in type and size of tubes.

EFFECTIVE PANEL CONDUCTANCE

If the effective conductance of the panel, C_e, is defined as the downward heat flow in Btu per (hour) (square foot) (Fahrenheit degree difference between

average tube temperature and average ceiling surface temperature), then $C_e = q_d/\Delta T$, or by Equation 1

$$C_e = q_d/(aq_d + bq_u)$$

or

$$C_e = 1/[a + b(q_u/q_d)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In Fig. 6, values of effective conductance C_e, calculated from average values of a and b, are plotted against the ratio q_u/q_d for panel No. 1. Experimentally

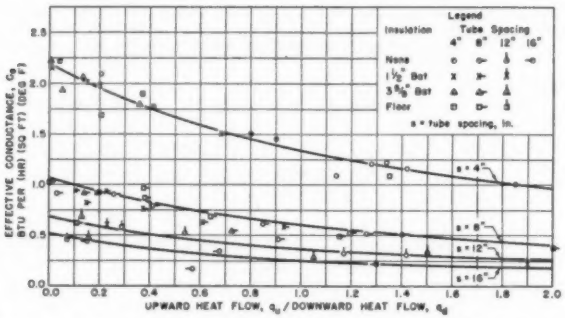

FIG. 6. COMPUTED CURVES AND EXPERIMENTAL VALUES OF EFFECTIVE CONDUCTANCE: PANEL NO. 1

determined points for different conditions of insulation are plotted to show their agreement with the averaged values. The curves of effective conductance shown in Fig. 6 are almost identical with those obtained for panel No. 3, and may be

TABLE 2—VALUES OF EFFECTIVE CONDUCTANCE, $q_d/\Delta T$

PANEL NO.	4 IN. TUBE SPACING			8 IN. TUBE SPACING		
	$\frac{q_u}{q_d} = 0.1$	$\frac{q_u}{q_d} = 0.4$	$\frac{q_u}{q_d} = 0.7$	$\frac{q_u}{q_d} = 0.1$	$\frac{q_u}{q_d} = 0.4$	$\frac{q_u}{q_d} = 0.7$
1	2.1	1.75	1.5	1.00	0.85	0.70
2	4.3	3.65	3.2	1.55	1.30	1.10
3	2.1	1.8	1.6	1.00	0.85	0.70
4	3.7	3.4	3.1	1.40	1.20	1.00

considered typical for panels having partially embedded tubes located above metal lath.

Fig. 7 shows curves of effective conductance C_e *vs.* the ratio q_u/q_d for panel No. 2 (tubes below metal lath). These curves also may be used for panel No. 4 (tubes below gypsum lath) without serious error.

Table 2 gives a comparison of the effective conductances for the four panels.

Here, again, it will be observed that the values for panels Nos. 1 and 3 are almost identical. It is also interesting to note that the values for panels 1 and 3, having partially embedded tubes above metal lath, are much lower than for panels 2 and 4 having tubes completely embedded.

EFFECT OF BACK INSULATION

The effect of heat loss from the back of the panel is also illustrated by Figs. 6 and 7. It will be noted that the effective conductance increases as the proportion of back heat loss or the ratio q_u/q_d decreases.

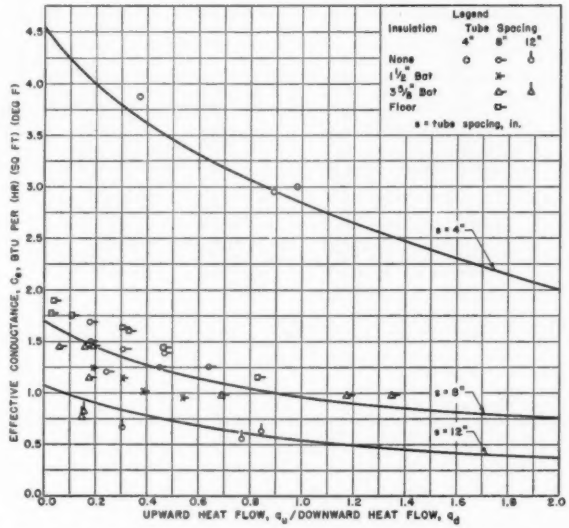

FIG. 7. COMPUTED CURVES AND EXPERIMENTAL VALUES OF EFFECTIVE CONDUCTANCE: PANEL NO. 2

To determine the effect of panel insulation, a special series of tests was made on panel No. 1. In these tests, the absorber and tube temperatures were held constant, while upward heat flow was varied by changing the insulation on the back of the panel. Test results are shown by the bar charts of Fig. 8. From the figure, it is evident that the upward heat flow, q_u, is materially reduced and the downward heat flow, q_d, is increased by back insulation. Thus, with an insulated panel a given rate of downward heat transfer may be obtained with a lower tube temperature than would be required with an uninsulated panel.

Tests with back insulation applied to panels 2, 3, and 4 produced approximately the same results as obtained for panel No. 1. The effects of back insulation, as shown in Fig. 8, may therefore be considered typical for all plaster panels.

A comparison of test data indicated that when both were operated without back insulation, the back losses from panel No. 3 were approximately 25 percent

higher than for panel No. 1. The greater loss was attributed to the larger area of bare pipe surface on panel No. 3, and the fact that its emissivity was considerably higher than for the non-ferrous tubes on panel No. 1. This difference in back losses, however, disappeared as soon as insulation was applied to the

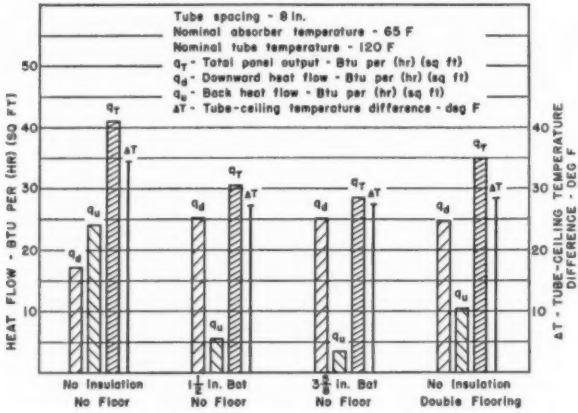
Tube spacing - 8 in.
Nominal absorber temperature - 65 F
Nominal tube temperature - 120 F
q_T - Total panel output - Btu per (hr) (sq ft)
q_d - Downward heat flow - Btu per (hr) (sq ft)
q_u - Back heat flow - Btu per (hr) (sq ft)
ΔT - Tube-ceiling temperature difference - deg F
HEAT FLOW - BTU PER (HR) (SQ FT)
ΔT - TUBE-CEILING TEMPERATURE DIFFERENCE - DEG F
No Insulation No Floor
1½ In. Bat No Floor
3⅝ In. Bat No Floor
No Insulation Double Flooring

FIG. 8. EFFECT OF BACK INSULATION ON PANEL PERFORMANCE: PANEL NO. 1

back of the panels. Since good practice requires the use of insulation, there would appear to be no preference for either type of tube on this particular basis.

EFFECT OF BACK-PLASTERING

The data in Table 3 have been arranged to show the effect of back-plastering. All tests were made with panel No. 1 without back insulation, and with 8 in. tube spacing. A comparison of the results of two pairs of comparable tests indi-

TABLE 3—DATA SHOWING EFFECT OF BACK PLASTERING ON PANEL NO 1. NO BACK INSULATION, 8 IN. TUBE SPACING

	NO BACK PLASTER	BACK PLASTER	NO BACK PLASTER	BACK PLASTER
Downward Heat Flow, q_d Btu/(hr)(sq ft)	17.2	23.3		20.1
Back Heat Flow, q_u Btu/(hr) (sq ft)	24.0	32.5	15.1	26.0
Tube to Avg. Ceiling Temp. Diff., F Deg	34.3	29.5	28.7	24.5
q_u/q_d	1.39	1.39	1.26	1.29
Effective Conductance, Btu/(hr) (sq ft) (F deg ΔT)	0.50	0.79 (0.75)[a]	0.52	0.82 (0.80)[a]
Upper Heat Absorber Temp. F Deg	65.1	64.7	76.4	75.2
Lower Heat Absorber Temp. F Deg	65.5	64.7	75.6	74.7

[a] Values for panel No. 2 for the same ratio of q_u/q_d and the same tube spacing.

cates that back-plastering caused an increase in heat flow from both sides of the panel, but that the ratio of the heat flow rates did not change appreciably.

It will be noted that the effective conductance of the panel was increased appreciably by the application of back-plaster. In both pairs of tests the effec-

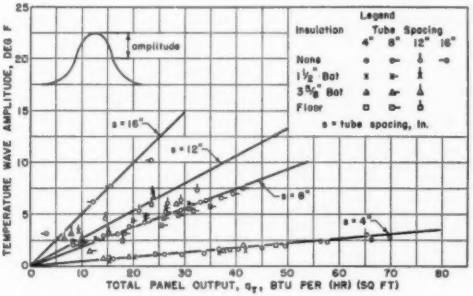

FIG. 9. TEMPERATURE WAVE AMPLITUDE *vs.* TOTAL PANEL OUTPUT: PANEL NO. 1

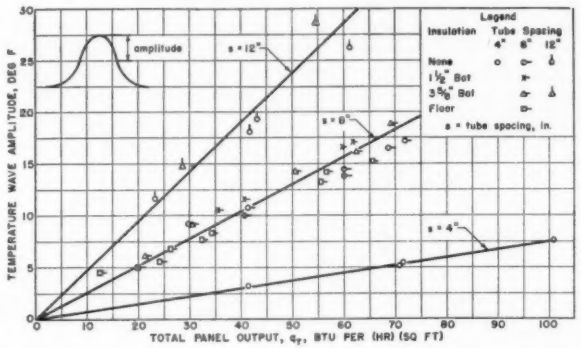

FIG. 10. TEMPERATURE WAVE AMPLITUDE *vs.* TOTAL PANEL OUTPUT: PANEL NO. 2

tive conductance for the back-plastered panel compared favorably with that determined for panel No. 2 under similar conditions. Therefore, a panel with tubes above metal lath, which is thoroughly back-plastered, may be expected to perform essentially the same as one in which the tubes are fastened below metal lath.

SURFACE TEMPERATURE WAVE

A curve of the surface temperature of a plaster panel along a line perpendicular to the tubes has a regular wave form, showing highest temperatures

immediately below the tubes and the lowest temperatures midway between the tubes. The amplitude of the wave increases with the tube spacing and total heat output, and is dependent somewhat upon the type of panel construction.

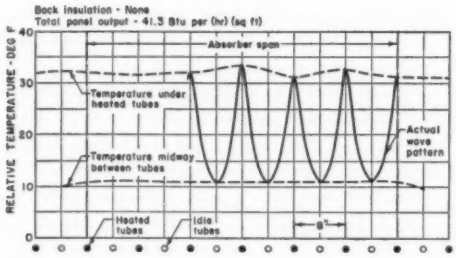

FIG. 11. RELATIVE CEILING SURFACE TEMPERATURES UNDER AND MIDWAY BETWEEN HEATED TUBES FOR PANEL NO. 2 (TUBES BELOW METAL LATH)

A correlation of the temperature wave amplitude with tube spacing and total panel output for panels Nos. 1 and 2 is presented in Figs. 9 and 10, respectively. It is evident from the test points shown on the figures that the wave amplitude for a given panel output is independent of back insulation. Curves for panels

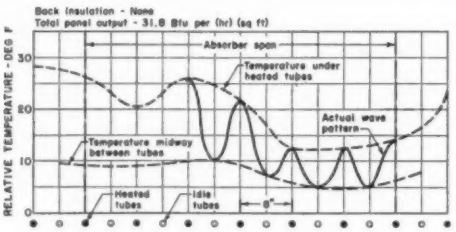

FIG. 12. RELATIVE CEILING SURFACE TEMPERATURES UNDER AND MIDWAY BETWEEN HEATED TUBES FOR PANEL NO. 1 (TUBES ABOVE METAL LATH)

Nos. 3 and 4 were fairly similar to those shown for panels Nos. 1 and 2, respectively.

The tubes in panels Nos. 2 and 4 were thoroughly embedded in plaster, and the heat transfer from tube to plaster was fairly uniform as illustrated in Fig. 11 by the curve showing the regular surface temperature wave for a typical test on panel No. 2.

In contrast to the regular temperature wave form shown for panel No. 2, Fig. 12 shows the variation in surface temperature and wave amplitude obtained for panel No. 1. These variations, observed with panels Nos. 1 and 3 were attributed to the non-uniform embedment of the tubes located above the metal lath. The portion of each tube of panel No. 1 in contact with the plaster was approximately the lower half (see Fig. 13). The pipe surface area in contact with the plaster in panel No. 3 was considerably less than 50 percent and this

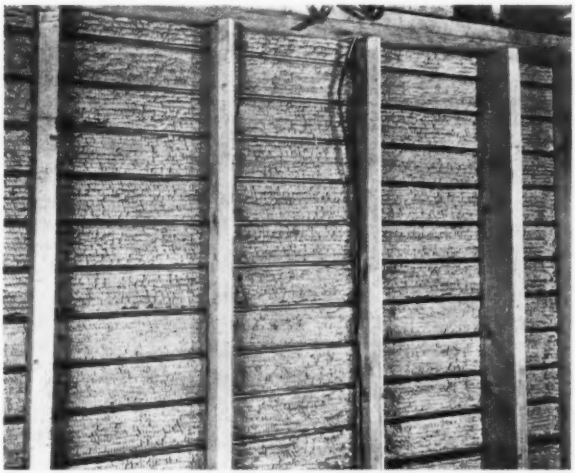

FIG. 13. BACK VIEW OF PANEL NO. 1

may be observed from Fig. 1. The actual area of contact, however, was as great, or greater, than in panel No. 1.

It should be emphasized that Figs. 11 and 12 are included only to show the character of the surface temperature wave pattern. The heat transfer rates for the tests shown in the figures are quite different, and the magnitude of wave amplitudes is not comparable.

CONCLUSIONS

1. The difference between the average tube temperature and the average panel surface temperature can be related to the heat output from the lower and upper panel surfaces by the empirical equation

$$\Delta T = aq_d + bq_u$$

2. Insulation on back of a plaster panel serves the following purposes:
   (*a*) It reduces upward heat flow.
   (*b*) It increases the average panel surface temperature for a given tube temperature.

(*c*) For a given downward heat flow it permits operation with a lower tube temperature than would be required if the panel were uninsulated.

3. A more uniform ceiling temperature *pattern* is obtained with completely embedded tubes. The temperature wave amplitude is greater, however, on the surface of the panel having tubes below the lath than it is on the surface of a panel having partially embedded tubes above the lath.

4. The back-plastering of a panel having tubes above metal lath increases its effective conductance to such an extent that its performance is essentially the same as that of a panel with tubes located below metal lath.

REFERENCES

1. Standard Specifications for Gypsum Plastering, obtainable from *The American Society for Testing Materials* or the *American Standards Association.*

2. A.S.H.V.E. RESEARCH REPORT No. 1387—Laboratory Studies on Heat Flow Within a Concrete Panel, by C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum and D. W. Locklin (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 175).

DISCUSSION

J. M. VANNIEUKERKEN, New York, N. Y. (WRITTEN): In their analysis of the test results on these panels the authors state that the coefficients a and b in Equation 1 were found to be almost independent of the amount of insulation back of a given panel construction and for a given tube spacing. Since no actual test values of these coefficients are tabulated in the report, while Figs. 4 and 5 merely present their values in the form of curves which were plotted on the assumption that they could be safely averaged, regardless of the amount or kind of back insulation used, it is impossible to check the accuracy of the foregoing statement.

The reason I am concerned about this statement which the authors make without furnishing supporting data for it is two-fold: First, because, without giving proof at the same time for a statement which is evidently open to challenge by anyone interested, the value of such a statement, and therefore the apparent results arrived at from the research work done, is seriously undermined; second, if the curves for a and b were used to evaluate the upward and downward heat flows of a panel and to that extent become part of a new design method or be embodied in already existing methods, there would be no certainty that these so-called improved methods would give more accurate results or be more reliable than those now in use.

Since ΔT is defined as the difference between the average tube temperature and the average panel surface temperature, it is suggested that, for clarity's sake, the word *surface* be inserted both in the bottom line of the legend and in the right-hand ordinate title of Fig. 8. It appears that the effectiveness of this chart would have been still further enhanced if it had been possible to make ΔT the same in all instances, including that of *No Insulation—No Floor.*

Although it is quite evident from Table 3 that the degree of embedment of the tubes in the plaster has considerable bearing on the uniformity of the panel surface temperature, as seems also illustrated by a comparison between Figs. 11 and 12, it would be interesting to know whether the fact that grid rather than sinuous coils were used might not have been a factor. If a corresponding chart were available for panel No. 3, this point might be cleared up. As it is, there can be no certainty that this lack of surface temperature uniformity was attributable altogether to non-uniformity in the plastering. In this connection it is of particular interest to note, as clearly shown in comparing Figs. 9 and 10, that the temperature wave amplitude in panels having

the tubes under the metal lath is for each output almost twice that of panels having the metal lath on the room side.

It must be evident to anyone, both from the irregularity of the amplitudes themselves and from the lack of uniformity of the temperature limit curves in Fig. 12 that the embedment of the tubes in panels 1 and 3 was not what it should have been, even if the plastering was done in conformity with the rules regarding acceptable workmanship. The statement in the report that in panel No. 3 considerably less than 50 percent of the pipe surface was in contact with the plaster seems clearly indicative of even the experimenters' beliefs that such embedment could not be considered the kind of workmanship needed to get the best results with ceiling type panels. Even though I am perfectly satisfied that the plastering of the panels done by the Laboratory was done as well as or even better than that in accepted commercial practice, lack of adequate, uniform embedment is, from a simple engineering standpoint, not *good* construction. I do not feel that results obtained from tests based on admittedly not-the-best-possible construction should be given the exalted position of scientifically established data on which dependable design methods should be reared.

Of course we cannot, in practice, expect perfect workmanship. However, to set an attainable goal we ought to know what can be obtained by perfect workmanship. By gauging from perfect workmanship a percentage of perfection which can reasonably be expected by ordinary commercial good workmanship, we can learn what to expect in practice. For that reason it would seem that tests should be made on additional plaster panels, as was suggested already during the meeting of the TAC on Panel Heating and Cooling at the 57th Annual Meeting in Philadelphia, 1951.

We are asked to be realistic in the matter, but are we asked to make allowances for indifferent workmanship when it comes to our war materials, jet planes, electronic devices, radar or gun controls? Although not in the class, perhaps, of the machinist and the fine-cabinet maker, the plasterer is also an artisan, who differs in pride of skill, or art, from these mentioned only in degree. If we, therefore, from scientific research and experiment have learned that a somewhat better class of workmanship is required in plastering where radiant heating panels are concerned, should we hesitate to appeal to the plasterer's pride or fail to set a somewhat higher standard in our plastering specifications for a radiant heating job? I believe that a little confidence in most people's desire to do a *good* job would benefit all of us, and in this case make for better radiant heating installations.

Another reason why it would seem desirable to have the Laboratory test additional plaster panels is that complaints have been coming in from certain areas of ceiling installations that have proved inadequate to heat the structures for which they were designed. In a number of cases this lack of capacity is believed to be attributable to the lack of understanding on the part of contractors and others of the effect of using plaster aggregates differing from those on which the design was based. This applies to the so-called light aggregates, such as perlite and various vermiculite mixtures which to a large extent are insulating and have K factors considerably under those given in THE GUIDE for plasters ordinarily used up to a few years ago. Establishment by the Society of the k factors of the most prevalent of these modern plasters and insistence by the Society that design specifications be rigidly followed, in regard to materials, construction, and starting of the installation, would help to solve some of the problems to which the art of radiant heating is still subject.

P. R. ACHENBACH, Washington, D. C.: The authors have presented some very interesting data on the heat flow within plaster panels. It answers some of the oft-asked questions about the relative location of the metal lath and the tubing and the differences between ferrous and non-ferrous tubing.

I would like to ask for a little clarification of one or two of the figures shown in the paper. It is stated that the temperature differences between tube and panel surface computed from the average values of a and b for 90 percent of the tests were

within 15 percent of the observed values. Do Figs. 6 and 7 include the tests that deviate by more than 15 percent? Was the variation in the amount of insulation the primary cause of the variation in the values of a and b? In developing Figs. 6 and 7 it is suggested that the variations of effective conductance with different amounts of insulation could be neglected. In discussing Fig. 8 and Table 2 it is shown that the effective conductance varies appreciably between insulated and uninsulated panels.

As near as I can read Fig. 8 the principal effect of insulating the back side of the panel is to reduce the upward heat flow. The downward heat flow was increased only about ½ Btu per hr (sq ft) or about 2 percent when 1½ in. of insulation was used as compared to the double flooring. Doubling the insulation only increased the downward heat flow another 2 percent approximately. Relating these variations to the ratio of q_u/q_d magnifies their significance when the changes in the absolute values are very small.

The condition represented in Fig. 8 where double wood flooring was used on the back side of the panel simulates the condition of a single panel delivering heat in both directions in a two-story installation. The upward heat flow was about 45 percent of the downward heat flow. This result corroborates data obtained at the National Bureau of Standards in our two-story panel-heating calorimeter. When a single wood floor is used on the back of the panel the upward heat flow was only about 60 percent of the downward heat flow. This is not usually a suitable division of the heat between a first and second floor in a two-story house, and indicates that a single frame panel with tubing in plaster on the lower side will probably not heat a first and second floor equally.

G. L. WIGGS, Montreal, Que.: This is a very interesting and valuable paper, but I feel that the average designer would have neither the time nor the inclination to go through the paper and do all the work involved to make practical use of it.

For example, the total heat flow is expressed as a function of the temperature difference between the average tube temperature and the average panel surface temperature in Equation 1. None of this information is normally available to the designer, but he is interested in the downward output from varying water temperatures. The total heat output of a panel is of secondary importance except in special cases, and is usually only considered in sizing the pipes and the boilers.

The interesting concept of effective conductance is proposed, but it varies with the ratio of upward heat transfer to downward heat transfer. If the upward heat transfer is not known, then no use can be made of the effective conductance. I believe that the value of the paper would be materially increased if it were possible to publish as an appendix: (*a*) Curves showing panel output downward to air at 70 deg with varying average water temperatures and with no insulation but with and without double floors, and (*b*) The same curves with the thicknesses of insulation used in the test.

R. J. ABRAMSON, Chicago, Ill.: We have in our section of the country several problems peculiar to our location that have to do with embedment depths of copper tubes in the ceilings and walls of radiant heated homes. With 1 in. thick plaster and an average embedment of ⅜ in., the following difficulties were experienced:

1. Plaster dried and cured at approximately 70 F. No hot water circulated in the copper tubes—dark surface markings at the tube locations. (This was eliminated by maintaining an 80 F panel surface temperature.)

2. After one season of use, the ceiling developed long cracks parallel to the tubes and approximately midway between the tubes. Increasing the depth of embedment to 1½ in. was partially successful. The use of 1 in. embedment with four (4) ounce muslin screeded into the white (finish) coat practically eliminated all troubles from this source.

We feel that our knowledge of heated plaster radiant panels is inadequate for intelligent and economical construction. We believe a sound research program based

on average construction practices and on panels as normally located to be of vital use to the designer.

E. S. HOWARTH.* New Kensington, Pa.: We are pleased to learn that the agreement between experimental and analytical results obtained in our paper on *Aluminum Ceiling Panels for Heating and Cooling* is of the same order, *i.e.*, ± 15 percent for 90 percent of the tests, as that reported in this work on plaster panels. The proximity of the upper and lower heat absorbers as shown in Fig. 3 raises a question concerning the possible influence of a temperature wave form, which might exist on the surfaces of the absorbers, on the observed temperature wave form for the test panel.

AUTHORS' CLOSURE (C. M. HUMPHREYS): Mr. vanNieukerken criticizes the statement that the coefficients a and b, which relate tube spacing to heat flow and tube-to-surface temperature difference, were found to be almost independent of the amount of back insulation applied to the panel. His criticism is well founded in that supporting data were not given.

Before presenting supporting data for the statement being questioned it seems appropriate to emphasize some of the difficulties involved. To be of real value, all of the data should be given, particularly in a problem of this nature for which a guiding analytical framework has not yet been developed. Since the results given in the paper are based upon over 180 individual tests, even the reduced data, let alone the raw experimental data would be voluminous.

As indicated in the paper, the curves of coefficients a and b which relate heat flow to tube-panel surface temperature difference represent average values for all panels, whether they are back insulated or not. The extent to which test results deviate from the mean curves can be illustrated in two ways. First, the panel conductance, which is the observed downward heat flow divided by the observed tube-panel surface temperature difference, can be compared with curves calculated from Equation 2 of the paper. Calculation of the curves involves selecting values of a and b from the curves of Figs. 4 and 5, and assigning convenient values to the ratio of upward to downward heat flow. This comparison was made in the paper in Figs. 6 and 7. The agreement of the test points appears to be satisfactory. Points representing insulated panels show no particular trend. The points given in Fig. 7 show more scatter than those in Fig. 6, but the points for uninsulated panels scatter to the same degree as those for insulated panels. The points for panels with double wood floors and 1½ in. bat insulation lie on opposite sides of the curve. One would expect them to be nearly the same, since the thermal resistances are of the same order of magnitude. This is probably due to differences in the experimental set up.

A second comparison can be made by computing values of ΔT by means of Equation 1 and the values of a and b as given by the empirical curves of Figs. 4 and 5 and comparing these results with experimentally observed values of ΔT. This has been done for a considerable portion of the data. Abstracts of this comparison are shown in Tables A and B.

For panel No. 1 with 4 in. tube spacing, Table A shows that the calculated and experimental values of ΔT are in almost exact agreement, for both the uninsulated panel and the panel insulated with 3⅝ in. bat insulation.

For the same panel with 12 in. tube spacing, Table B shows that calculated values are 5 percent above the experimental values for the uninsulated panel and 11 percent below for the panel with 3⅝ in. insulation. The deviations of the calculated ΔT's for panels with 1½ in. bat insulation are 2 and 4 percent below the experimental values, for 4 and 12 in. spacings respectively.

In the case of panel No. 2, with no insulation, calculated ΔT's for 4, 8, and 12 in. spacings are 6, 13 and 3 percent above the experimental values. Tests on a 3⅝ in.

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* Chief, Metal Working Division, Aluminum Research Laboratories, Aluminum Company of America.

bat insulated panel were made for 8 and 12 in. spacings only. ΔT's for these averaged 11 and 15 percent below the experimental values.

In the case of panel No. 3, with no insulation, calculated ΔT's for 4, 8 and 12 in. spacings were 8 percent below, 10 and 7 percent above, respectively. Tests on an insulated panel were made with 8 in. spacing only. ΔT's for these tests were 5 percent below the experimental ΔT's.

TABLE A—COMPARISON OF EXPERIMENTAL AND CALCULATED ΔT

Panel No. 1: 4" Spacing; $a = 0.44$, $b = 0.30$

No Insulation				3½ In. Bat			
		ΔT				ΔT	
q_d	q_u	Calc	Exp	q_d	q_u	Calc	Exp
25.0	20.0	17.0	16.4	37.1	0.2	16.9	16.7
16.2	20.7	13.3	13.3	13.6	0.7	6.2	6.9
5.6	24.0	9.7	9.9	14.0	5.1	7.7	7.7
7.5	8.6	5.9	6.7	36.7	4.9	17.6	17.5
18.5	7.7	10.4	10.4				
	Ave.	11.3	11.3		Ave.	12.1	12.2

Thus it appears that the curves of a and b shown in Figs. 4 and 5 give calculated values of ΔT for both insulated and uninsulated panels with practical accuracy. The

TABLE B—COMPARISON OF EXPERIMENTAL AND CALCULATED ΔT

Panel No. 1: 12" Spacing; $a = 1.38$, $b = 1.19$

No Insulation				3½ In. Bat			
		ΔT				ΔT	
q_d	q_u	Calc	Exp	q_d	q_u	Calc	Exp
12.9	19.7	41.2	38.3	21.3	2.7	32.6	31.1
5.3	24.0	35.9	32.7	3.9	4.1	10.3	13.8
8.7	12.4	26.8	27.5	5.7	0.9	8.9	11.9
17.5	9.5	35.4	33.1	20.4	−1.6	26.3	30.7
4.6	5.4	12.8	13.9				
	Ave.	30.4	29.1		Ave.	19.5	21.9

experimental results, however, indicate that ΔT's for the same tube spacing and heat flow rates run a little higher for insulated panels than for uninsulated panels, but the differences were not great in the tests made on the panels described in the paper.

It should be noted that the form of Equation 1 is such that ΔT depends upon the combined effect of a and b. For example, if the upward heat flow is small, as in an insulated panel, b can be varied considerably without appreciably affecting the calculated value of ΔT. The present curves result from repeated trial and error calculations to find the best fit for the data.

We agree with Mr. vanNieukerken that everything possible should be done to improve the quality of the plastering job on a ceiling panel. We do not agree that tests should be made on *perfect* panels to develop data on which to design *average panels*. In constructing the test panels at the Laboratory every effort was made to duplicate *good* field practice. In his discussion, Mr. vanNieukerken, agrees that the

"plastering . . . was done as well or even better than that in accepted commercial practice. . . ."

The average embedment of the tubes in panel No. 1 was approximately 50 percent, and it is doubtful if appreciably better tube embedment can be obtained without back-plastering. The non-uniform surface temperature wave shown in Fig. 12 is for this panel. While this lack of uniformity may be due in part to variations in tube embedment, the varying degree of contact between the tubes and the metal lath may have a very pronounced effect.

It may be desirable to test a panel constructed with vermiculite or similar insulating plaster aggregates. Certainly the heat transfer data developed for ordinary gypsum plaster cannot be applied to insulating plaster.

At best, the Society can do no more than suggest or recommend certain procedures for the design and construction of heating panels. It cannot insist that they be followed.

Answering Mr. Achenbach's questions in the order in which they were stated in his written discussion;

1. All of the test points, including those which deviated more than 15 percent from the curve values are shown in Figs. 6 and 7.

2. The values of coefficients a and b were found to be almost independent of the amount of back insulation used. As shown in Figs. 4 and 5, the values of a and b were somewhat different for each panel tested, but the most important factor in determining their magnitude was the tube spacing.

3. The curves of Figs. 6 and 7 were calculated by Equation 2 from the values of a and b taken from Figs. 4 and 5. As stated above, these values of a and b were found to be practically independent of the effect of back insulation. However, it does not follow that C_e, the effective conductance, is also independent of back insulation effect. In the laboratory, the ratio of upward to downward heat flow, q_u/q_d, could be varied by varying the absorber temperatures by using insulation, or by both methods. In practice, for a given installation and a given downward heat flow, the ratio q_u/q_d can be varied only by changing the thermal resistance of the heat path from the back of the panel. A low ratio of heat flow rates would require a well insulated panel. The curves of Figs. 6 and 7 definitely indicate an increase in effective conductance with a decreasing value of q_u/q_d. This is consistent with the results indicated in Fig. 8 and Table 2.

It is possible that Mr. Achenbach was confused by the experimentally determined points for various conditions of insulation which are plotted on Figs. 6 and 7. These were shown primarily to justify the method of analysis used, and to further justify the statement that the values of a and b are independent of insulation conditions.

Mr. Achenbach is correct in pointing out that the principal effect of back insulation is to reduce the upward heat flow. As stated in the conclusions of the paper, it also increases the average ceiling temperature for a given tube temperature, and in so doing, makes it possible to obtain a given downward heat flow with a lower tube temperature.

Mr. Wiggs is correct in pointing out that the data, as presented in the paper, are not in the most usable form. The paper as first written included a simplified design procedure, but the TAC on Panel Heating and Cooling felt that that step should be omitted. A design procedure, based on the Laboratory test, should be developed and made available for use as soon as practicable.

Actually, the application of the data presented in the paper is not as difficult as it may at first seem. The upward heat flow for a given condition of insulation can be readily approximated. Knowing this, and the desired rate of downward heat flow, the effective conductance can be determined from Figs. 6 and 7.

Mr. Abramson has touched on a subject which is of great interest to everyone who is designing or installing plaster panel systems. However, since the work at the

Laboratory was confined to a study of the thermal characteristics of plaster panels, we have no suggestions to offer on the streaking and cracking of plaster.

In answer to Mr. Howarth's question, the variation in temperature across the surface of the absorber was extremely small. The refrigeration plate used in the absorber construction had liquid passages located only ⅞ in. center to center. Below each plate was a sheet of 3/32 in. aluminum which further reduced temperature variations across the surface. It is therefore safe to say that the temperature wave observed on the plaster panel surface was not in any way influenced by variations existing on the absorber surfaces.

1427

No. 1427

OPTIMUM PANEL SURFACE DISTRIBUTION DETERMINED FROM HUMAN SHAPE FACTORS

By F. W. Hutchinson,* Berkeley, Calif., and Merl Baker,** Lexington, Ky.

EIGHT years ago, as part of a program of research sponsored by The American Society of Heating and Ventilating Engineers in cooperation with the University of California, a progress report[1] was presented giving experimentally determined human shape factors for seated and standing subjects in rooms with 8, 10, 12 ft heated or cooled ceilings. The present report, submitted long after conclusion of the cooperative panel heating project, supplements the original study by presenting data for a human subject with respect to a heated or cooled floor panel.

The two major applications of human shape factors are: (1) In determining that distribution of heating or cooling surface (in a floor or ceiling that is not 100 percent heated or cooled) for which the optimum uniformity of heating or cooling effect will be experienced by the occupants. (2) In determining the size of a heating or cooling panel for producing a localized region of greater warmth or of increased cooling effect; the importance of this second application is particularly great in the solution of industrial problems such as occur in production lines or in large structures having low density of occupancy.

Theory

In any heated or cooled structure, under conditions of steady state, the only condition which can lead to variation in the feeling of warmth experienced by an occupant is change in the geometrical position of that occupant with respect to the surrounding surfaces. Thus, due either to movement from place-to-place in the room or to alteration of the geometry as by standing, sitting, or reclining, the radiant exchange between the occupant and the surroundings may be subject to variation. Maintenance of uniform warmth is a prerequisite to comfort, hence the designer must seek to minimize changes in net radiant transfer that occur due either to movement or to change in position. Analysis of such changes requires evaluation of the net transfer to-or-from the occupant.

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Portland, Ore., July 1951.

The geometry of an occupant with respect to his surroundings is expressed in terms of a shape factor which may be defined by Equation 1,

$$F_{os} = (1/A_s) \int_{A_o} \int_{A_s} \frac{\cos \phi_o \cos \phi_s \, dA_o \, dA_s}{\pi (r_{os})^2} \quad \ldots\ldots\ldots \quad (1)$$

where,

F_{os} = shape factor of the occupant with respect to energy received from surface A_s, or fraction of energy emitted by A_s which is received by the occupant.
r_{os} = distance between center of surfaces A_o and A_s.
ϕ_o = angle between r_{os} and normal to center of A_o.
ϕ_s = angle between r_{os} and normal to center of A_s.

The emissivity of an average clothed person closely approaches unity (hence the absorptivity is likewise close to unity) so the net transfer rate, by radiation, between an occupant and any surface which he *sees* is given by Equation 2,

$$q_{os} = 0.172 \, (e_s) \, (F_{os}) \, (A_s) \, [(T_s/100)^4 - (T_o/100)^4] \quad \ldots\ldots\ldots \quad (2)$$

where

q_{os} = net radiant transfer, Btu per hour from A_s to occupant.
e_s = emissivity of the surface A_s.
A_s = area of the surface which the occupant *sees*.
T_s = absolute surface temperature of A_s.
T_o = absolute surface temperature of the occupant.

PROCEDURE

Because of the complex curves and projections of the human body it is not possible by any convenient mathematical simplification to evaluate accurately the double-integral equation which defines the shape factor of an occupant with respect to a panel. The aesthetically pleasing diversity of human shapes is responsible for insurmountable technical difficulties with respect to the rational evaluation of either individual shape factors or an average human shape factor. The investigation reported in this paper was intended to provide accurate experimentally determined shape factors for an *average* person with respect to energy emitted by areas in the floor at varying distances from the point at which the subject is standing. As a check on the applicability of *average man* factors to markedly *non-average* individuals the investigators were fortunate in securing the anonymously-given services of some 10 subjects who collectively represented the extremes (for 95 percent of the population) of variation in height, width, depth, position of center of gravity, and maximum of local projection of the clothed human figure in profile and in full-face view. Since the shape factors of all but one of the *boundary* subjects were within plus or minus 4 percent agreement with the shape factor of the average subject (for floor areas more than 5 ft 6 in. out from the position of the standing subject) it appears that the shape factors given here can be used with acceptable accuracy for any person whose figure is close enough to the accepted norm to permit escaping marked public attention.

The experimental procedure followed in collecting data for this paper was the same as that used in the earlier investigation[1] of human shape factors with respect to wall panels and to ceiling panels. In brief summary, the test subject was a clothed dummy representing a 5 ft 10 in. man weighing 165 lb; the dummy was dressed in a two-piece suit, remained in standing position, and had both arms at its sides. Shape factors were determined experimentally by means

[1] Exponent numerals refer to Bibliography.

of a mechanical integrator of the type proposed by Hottel† and developed by Boelter†† and associates. Since the earlier tests showed that full-front shape factors agree closely with full-back factors and that front semi-profile factors differ but little from rear semi-profile shape factors the present investigation was limited to a determination of full-face and of front semi-profile values. The data for these two positions can then be readily interpolated and extrapolated to permit development of a complete *field* for each of the four identical quadrants which make up the floor region within an 18 ft radius of the subject.

Since the exchange rate, for a fixed position of the occupant, is the sum of the exchange rates for all unit areas of the panel, the problem resolves itself into a determination of the exchange variation between a fixed unit area (or infinitesimal area) of panel and a moving occupant. More simply, the experimental data can be collected for a system in which the occupant retains a fixed position and the element of panel is moved.

As used in this research the mechanical integrator was placed directly over the point selected to represent the center of the elementary panel area. From this position a beam of light from the integrator was directed around the outline of the subject. For each such determination the integrator drew a closed curve on a piece of paper. The area within the curve was determined with a planimeter and divided by an instrument constant to obtain the shape factor of the subject with respect to energy emission from the point source located at the center of the elementary panel area. Shape factors were obtained at 1-ft intervals from the dummy to a maximum distance of 18 ft. Re-determinations were made at each point at least three times and never less than the number of times required to assure reproducibility of results.

NEW EXPERIMENTAL RESULTS

Floor Panels. Fig. 1 presents the two basic curves which summarize the new experimental data. The abscissa can be considered as representing either the distance from a fixed subject to various elements of floor area or the distance from a fixed element of floor area to a subject who is moving around in the room. The ordinate is numerically equal to the fraction of energy emitted by a diffuse floor surface which will be intercepted by the standing subject; the shape factor assists in determining irradiation of the subject, but not energy reception, since the latter depends on the absorbtivity of the receiving surface. With very few exceptions the materials ordinarily used for clothing have emissivities substantially greater than 90 percent, hence, for practical engineering calculations it is usually sufficiently accurate to assume that all long-wave length irradiation is absorbed by the subject.

Examination of Fig. 1 shows that maximum irradiation occurs from areas 2 ft out from the subject and that beyond this distance there is a very rapid falling off of the shape factor. Five feet out from the subject the fraction of energy leaving the floor which strikes the occupant is under 2 percent and 7 ft out, this fraction has decreased to less than 1 percent. Beyond 10 ft less than one-half of 1 percent of the energy emitted by a warm floor will provide irradiation of the occupant. For an occupant remaining in a fixed position—as in some

† Radiant Heat Transmission, by H. C. Hottel, (*Mechanical Engineering*, Vol. 52, 1930).

†† A Mechanical Integrator for the Determination of the Illumination from Diffuse Surface Sources, by V. H. Cherry, D. D. Davis, L. M. K. Boelter, (*I.E.S. Transactions*, 1939).

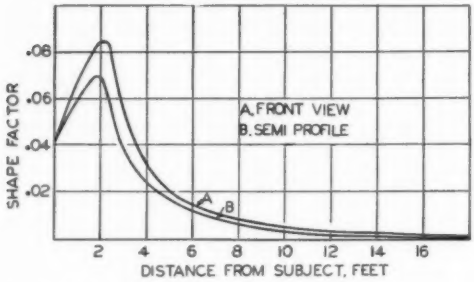

FIG. 1. FRACTION OF ENERGY REACHING A STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA

kinds of factory work—it is evident that maximum effectiveness of local heating (or cooling) would be attained by providing a heated *ring* of floor surface having an inside diameter of approximately 3 ft 6 in. and an outside diameter of 4 ft 6 in.; the average direct radiant transfer from this ring to the occupant would be of the order of 7 percent hence, per unit area, the ring would be some 700 to 1000 percent more effective than uniform heating of the entire floor area.

APPLICATION OF FLOOR DATA

The data of Fig. 1 can be used directly to determine the shape factor of any floor panel of any size or shape with respect to a standing subject in any position, either on or off of the panel. The procedure is to divide the panel into equal areas (preferably not larger than 2 by 2 ft on the sides) and determine the distance in feet from the subject to the center of each such area; then reference to Fig. 1 will give the shape factor of the subject with respect to energy received from each of the areas and the *average* of the individual area shape factors will be equal to the shape factor of the subject with respect to energy received from the entire panel. By the same method the variation in energy reception from a given panel as a function of the position of the occupant can be readily investigated.

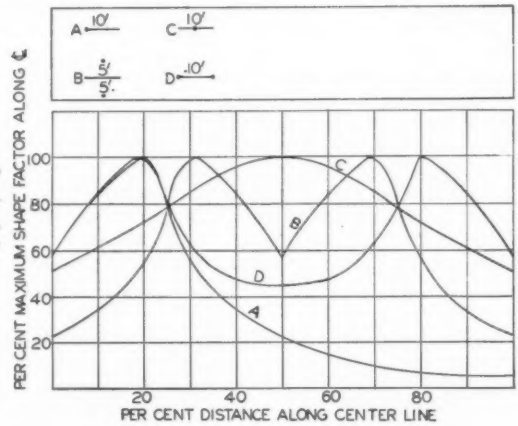

FIG. 2. COMPARISON OF SHAPE FACTOR VARIATIONS FOR THE PANEL ARRANGEMENTS SHOWN

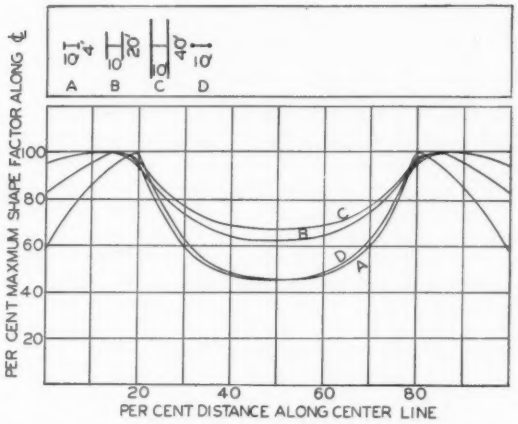

FIG. 3. COMPARISON OF SHAPE FACTOR VARIATIONS FOR THE PANEL ARRANGEMENTS SHOWN

A knowledge of human shape factors is necessary in the design of local radiant systems for either heating or cooling, and in the determination of optimum distribution of heated or cooled surface in a room for which the total load is insufficient to justify 100 percent coverage. The first of these problems requires special study for each individual application and the procedure is identical with that described in the preceding paragraph. The second problem more readily lends itself to generalized solution hence will be given more detailed consideration in the following section.

OPTIMUM SURFACE DISTRIBUTION

Floor Panels. In any panel heated or cooled enclosure the designer must establish environmental conditions such that uniform warmth will be experienced by the occupant irrespective of his position in the room. This condition requires

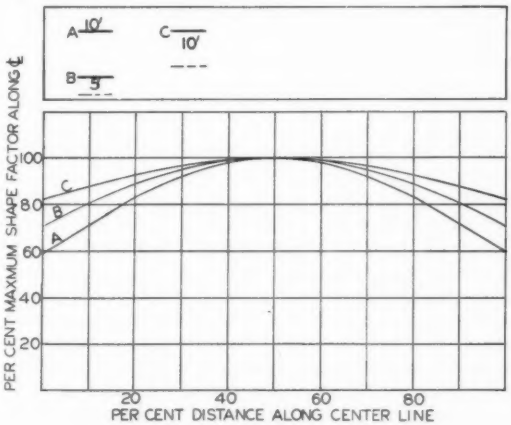

FIG. 4. COMPARISON OF SHAPE FACTOR VARIATIONS FOR THE PANEL ARRANGEMENTS SHOWN

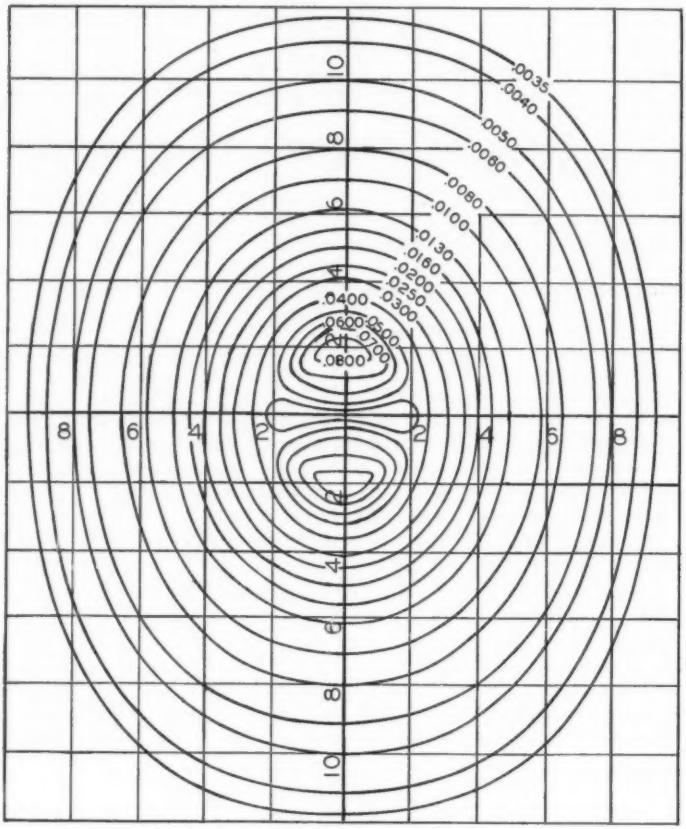

FIG. 5. FLOOR PANEL FOR STANDING SUBJECT

minimum variation in the direct transfer of radiant energy between the occupant and the primary heating or cooling surface. Referring to Fig. 2, Curve A shows the percentage of maximum shape factor that would exist as an occupant moved backward or forward along a line 10 ft out from a small element of primary area; when 7 ft out from the area the irradiation rate would be 10 percent of that corresponding to a distance of 2 ft out.

Curve B of Fig. 2 shows variation over the same 10 ft line, but with small primary areas located 5 ft out from the center of the line on either side; in this case substantial improvement in uniformity over the conditions of Case A has been achieved. Curve C is for the unit area located at the center of the 10 ft line and again shows improvement over the arrangements of Curves A and B. Curve D, for unit areas at each end of the line, is practically equivalent to Curve

C as far as maximum variation goes, but is less effective because of the rapid change in shape factor at a distance 2 ft in from each end of the line.

Fig. 3 provides an extension of the condition shown by Curve D of Fig. 2 (which curve is reproduced, for purposes of comparison as Curve D of Fig. 3). Curves A, B, and C of Fig. 3 are for line panels of 4, 20, and 40 ft, respectively, centered at either end of the 10 ft line and normal to it. Reference to these three curves will show that both the center and end *droops* of the shape factor variation decrease with increasing length of the line panel. For lengths beyond 20 ft, however, the influence of panel length on droop is very much reduced.

Fig. 4 shows the variation for the case of a line panel parallel with the path of the occupant and either superimposed on it (Curve A) or a distance of 5 ft out (Curve B) or 10 ft out (Curve C). Once again it is immediately evident that uniformity of direct radiant transfer increases as the panel moves farther away from the subject. Although Fig. 4 does not show curves for panel distances greater than 10 ft out, a simple analysis by the method already described will demonstrate that there is relatively small improvement in uniformity for panel distances appreciably greater than 10 ft.

Comparing Curves B of Fig. 3 and B of Fig. 4, it is noted that these two types of panel distribution complement one another. From this fact it appears that an excellent degree of uniformity of primary radiant exchange would be attained in a room 10 by 10 ft if the heated section of the floor were limited to a band running around the periphery of the room. For a room 20 by 20 ft excellent distribution would be achieved by a peripheral panel combined with two center line cross-panels which would make the distribution equivalent to that in four adjacent 10 by 10 ft rooms. The present analysis does not provide sufficient data to warrant the conclusion that 10 ft spacing of the strip panels is optimum, but the reader can readily verify that 10 ft is more effective than any lesser spacing and is but little less effective than any greater spacing.

For the great majority of rooms that are heated or cooled by means of floor panels the required ratio of panel surface to total floor surface is so great that no attempt can be made to achieve uniformity of direct transfer by selective distribution of panel surface. The possibility does exist, however, of improving distribution in such cases by designing for a higher surface temperature in the 2- or 3-ft wide peripheral section; with sinuous coil or grid-coil panels this can be done by spacing the pipe or tube more closely in the peripheral section. Aside from increased uniformity of direct transfer such peripheral emphasis is also desirable, particularly near outside walls, as a means of offsetting coldwall effect and thereby improving uniformity of the overall heat balance (as contrasted with uniformity of direct primary transfer).

INTEGRATION OF ALL EXPERIMENTAL RESULTS

Fig. 5 obtained by interpolation and extrapolation of the data from Fig. 1, visualizes the occupant as standing at the center of coordinates and facing in the positive direction of the y-axis. Each curved line of the figure represents a locus of points on the floor around the occupant for which the shape factor has constant value; such iso-shape factor lines permit direct visualization of the fraction of energy emitted by any point on the floor and striking the occupant. Thus from a point on the floor 2 ft to the right and 2 ft ahead of the occupant 4 percent of the emitted energy would be received by the occupant; the remaining 96 per-

cent of emitted energy would miss the occupant and be received by interior surfaces of the room. Although values of the shape factor shown on the figure are specifically determined for point sources they are nonetheless applicable to any finite area of panel over which the average shape factor approaches with adequate engineering accuracy the value of the shape factor at the center of that area. To determine the shape factor of occupant with respect to any large area (as a floor panel) the panel can be sketched on Fig. 5 in its proper position with respect to the occupant and the overall shape factor then determined as the average (*not* the sum) of shape factors of selected unit areas of the large panel.

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2. Experimental Evaluation of Human Shape Factors with Respect to Floor Areas, by F. W. Hutchinson (*A.S.M.E. Transactions*, August, 1950).

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DISCUSSION

R. G. VANDERWEIL, Boston, Mass. (WRITTEN): Human shape factors with respect to floor areas together with those in respect to wall and ceiling areas, will supply the designing engineer with the data required for the solution of a number of special heating and cooling problems. It will help in the design of cooling systems for spaces such as rooms containing great numbers of business machines or other heat generating equipment; even more important, it will enable the engineer to design the most economical heating systems for large industrial plants where aisle locations are fixed for present as well as for future operation. It is conceivable that by the use of properly located floor panels in such buildings, large fuel savings may be obtained.

With reference to the application of the human shape factor to the design of conventional room heating systems, I should like to express appreciation of the authors' remarks on the great majority of heated rooms. It appears that in the 10 by 10 ft room a shallow circumferential coil presents an ideal solution of the heating problem—if none of the walls are outside walls and the ceiling or roof loss is fairly constant through all points of the ceiling.

However, as the authors point out, when one of the walls of this room is an outside wall, and therefore had a considerably lower surface temperature than other room surfaces, it will be necessary to concentrate the heat output of the floor or ceiling panel in the area adjacent to this wall.

We should remember that maximum comfort in a room, in regard to heat interchange by radiation, will be maintained if the difference between heat loss and heat gain per square inch of human body—front, back or side—remains constant no matter where the occupants may be located within the room.

I should not necessarily desire to steadily occupy a room designed according to such principles for fear that I may grow stale and miss the refreshing effect of a few cold spots. However, extreme variations of the heat balance of a human body within a room should be avoided. The heating designer became aware of these principles many years ago when he removed the radiator from the inside wall and replaced it by radiation under the window.

A complete analysis of radiation conditions within a room would have to consider radiation influx from the heated panel superimposed over the radiation loss of the human body to the cold outside wall and glass.

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BAROMETRIC DRAFT-CONTROL ANALYSIS

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CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

DRAFT control is an important subject relating to: (1) fuel economy, (2) safety, (3) comfort from even heating rates, and (4) design of appliances and chimneys.

The Research Technical Advisory Committee on Combustion has recommended the barometric draft control as needing research study. A survey of the practical aspects of design and performance[1] led to a program being outlined with the objectives: (1) to determine representative basic data needed to improve barometric draft-control design and performance, (2) to aid effective designs and installations by an exposition of physical principles, and (3) to work toward possible standards of value to the industries concerned. The initial steps reported here comprise: (1) *damper mechanics,* and (2) *damper through-flow characteristics.*

DEFINITION

A barometric draft control is defined as an automatic, adjustable, or factory-set, self-contained, draft-actuated device the purpose of which is: (1) to prevent excessive draft on a fuel-burning appliance, and (2) to make the appliance draft nearly independent of chimney draft over the normal range of operation. This device is to embody moving parts which operate by opening a by-pass from the

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

surroundings into the flue pipe, or by restricting the flow in the flue pipe, or both, in accord with the magnitude of the appliance draft.

The barometric draft control is generally applied to coal or oil burning installations and is not to be confused with draft hoods used in gas installations. (Instances also are known where these controls have been employed in ventilation systems.)

SCOPE OF INVESTIGATION

Performance rating or selection of a barometric draft control requires either: (1) an actual test of whatever complete draft system is involved, or (2) availa-

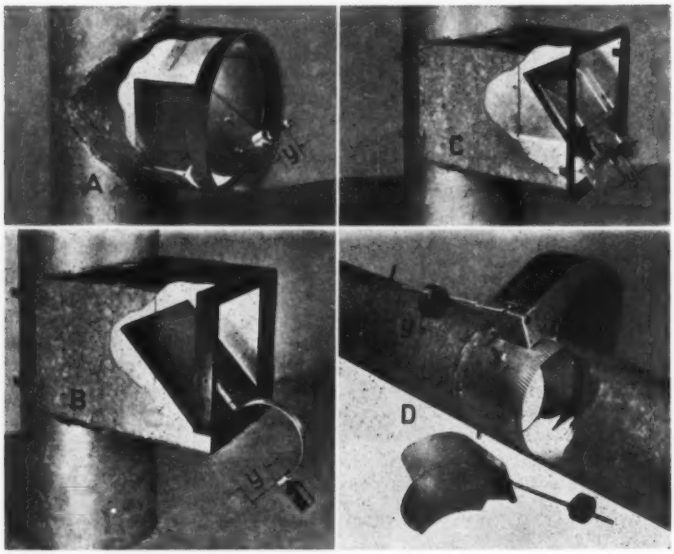


FIG. 1. BAROMETRIC DRAFT CONTROLS STUDIED

bility of proper fundamental data of such a nature that a trustworthy performance prediction may be made. The program outlined is dedicated to the second alternative, which calls for the synthesis of system performance from component-element data.

To obtain exemplifying data on physical principles, four types of barometric draft controls were selected; these are illustrated in Fig. 1 and described in Table 1. Only units in an unused condition were studied. Effects of service deterioration were beyond the scope of the first part of the program. Furthermore, only static or equilibrium behavior was investigated; problems of dynamic behavior were left untouched.

Fig. 2 introduces and orients the constituent aspects of the present studies. Each topic is subject to planned design control with the aid of a few careful experiments.

DAMPER MECHANICS

Referring to Fig. 2, damper mechanics comprises analysis of design details, the actuating forces producing damper motion, and whatever positioning characteristics may be desired.

An installed barometric draft control positions itself through force action and thereby produces flow regulation. The damper position is the common factor in tying damper mechanics to flow phenomena.

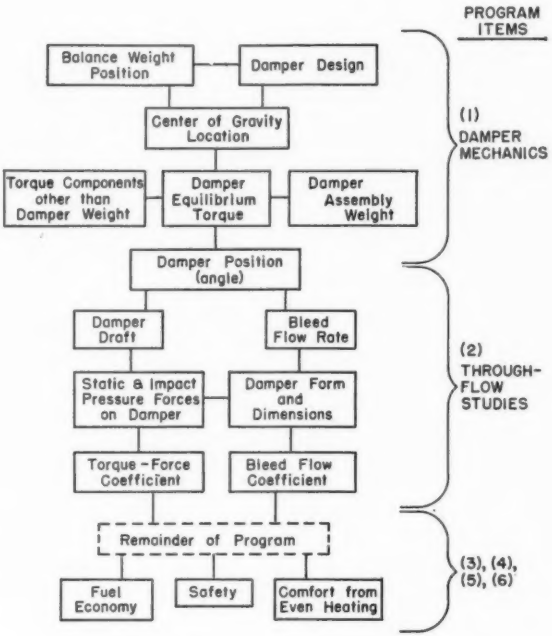


FIG. 2. INTER-RELATIONSHIPS STUDIED

Rotation results from torque action. As long as the rotating damper assembly operates as a rigid body, the torque required to balance the assembly at any angular displacement from free rest is defined, following Fig. 3, as

$$L = Wr \sin \phi \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

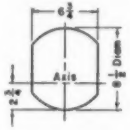
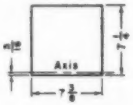
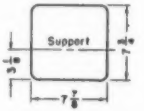
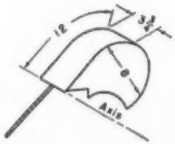
where

L = balancing torque, pound-inches.
W = weight of all rotating parts, pounds.
r = distance between center of gravity and axis of rotation, inches.
ϕ = angular displacement from free rest.

The center-of-gravity location is of outstanding operating importance. A fully-closed damper generally is not in the free-rest position. Establishing the

TABLE 1—FEATURES OF BAROMETRIC DRAFT-CONTROL DAMPERS TESTED

(All dampers were for 8-in. nominal duct size.)

DRAFT CONTROL	WEIGHT ALL ROTATING PARTS, LB	WEIGHT OF BALANCE WEIGHT, LB	NATURE OF ROTATION AXIS SUPPORT	BALANCE WEIGHT ADJUSTMENT	FLOW PATH FEATURES	DAMPER FACE DIMENSIONS, INCHES
A	1.52	0.166	Damper fixed to axis which rolls along horizontal track.	Along threaded bolt at angle to damper face, with fixed counterweight behind damper.	Air admitted both above and below axis of rotation. Side guide vanes and turned-back damper edges employed.	
B	1.29	0.419	Damper rotates on pins attached to frame close to bottom edge.	Along slide arm suspended below and normal to the damper face.	Air passes around upper and side edges of inclined damper into rectangular duct section.	
C	0.404	0.083	Damper supported by thin sheet springs which bend along sector arm during rotation.	Along threaded bolt set normal to damper face.	Air admitted both above and below axis of rotation. Damper edges are sharp.	
D	2.01	0.69	Damper rotates on pins attached to frame.	Along threaded balance rod, extending out on opposite side of axis of rotation from damper itself.	Combines partial throttling of through-run (appliance exhaust) flow with admission of some air from surroundings through clearance space between damper and housing.	

center of gravity, therefore, requires either: (1) tests, or (2) a design analysis from a drafting-table layout. The first of these approaches was followed in the present instance and results are contained in Appendix A.

A review of Fig. 4 yields practical interpretations of center-of-gravity behavior for a rigid rotating assembly:

1. In a position of free rest (*i.e.*, no external torque) the center of gravity will be at point A_1, vertically below the axis.

2. The torque will be a maximum at a rotation of 90 deg from A_1 to A_2.

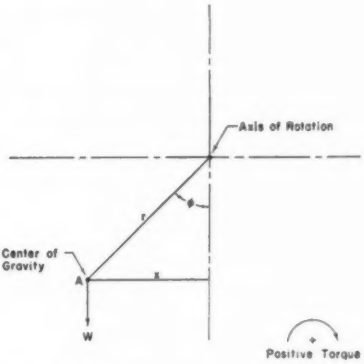


FIG. 3. DAMPER-WEIGHT TORQUE ABOUT AXIS

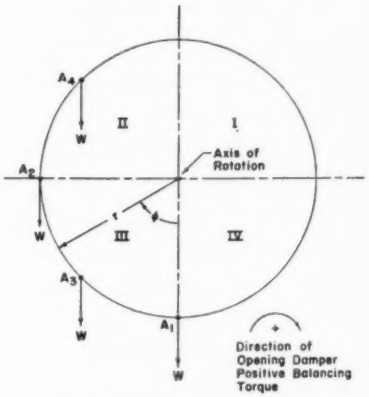


FIG. 4. QUADRANT NOTATION FOR CENTER-OF-GRAVITY LOCATION

3. For any center-of-gravity position in quadrants III or IV, as A_3, the torque equilibrium is inherently stable.

4. For any center-of-gravity position in quadrants I or II, as A_4, the torque equilibrium may easily be unstable.

Torque Data: Balancing-torque data were obtained by applying a known external torque to each damper with the aid of an apparatus which permitted either the damper frame, the damper blade, or both, to assume any rotational position. (These tests were made mechanically with the damper frame mounted separate from any duct attachment.) Determinations are summarized in Figs. 5, 6, 7 and 8. The solid lines in each instance are sine curves drawn to give the best fit for the test points, except for Fig. 7, Damper C, where a sine curve would not apply because of the torque contributed by the spring supports. The dashed curves in Fig 6 are interpolations.

In an actual draft-system installation, a pressure difference across the damper produces a torque. The damper moves until the pressure-caused torque (herein called the *balancing torque*) comes into balance with the restoring torque which arises from the weight of the rotating assembly, plus the action of springs or similar external attachments. Torque equilibrium requires that the balancing and restoring torques have the same magnitude, but opposite directions.

Referring back to Fig. 4 for the free-swinging dampers, the points of zero and of maximum balancing torque represent, respectively, locations A_1 and A_2 for the center of gravity when they are separated by 90 angular degrees in a sequence of torque increasing with angle. One hundred and eighty angular degrees from free rest is also a position of zero torque, but here the equilibrium is unstable except for aid from possible frictional restraint.

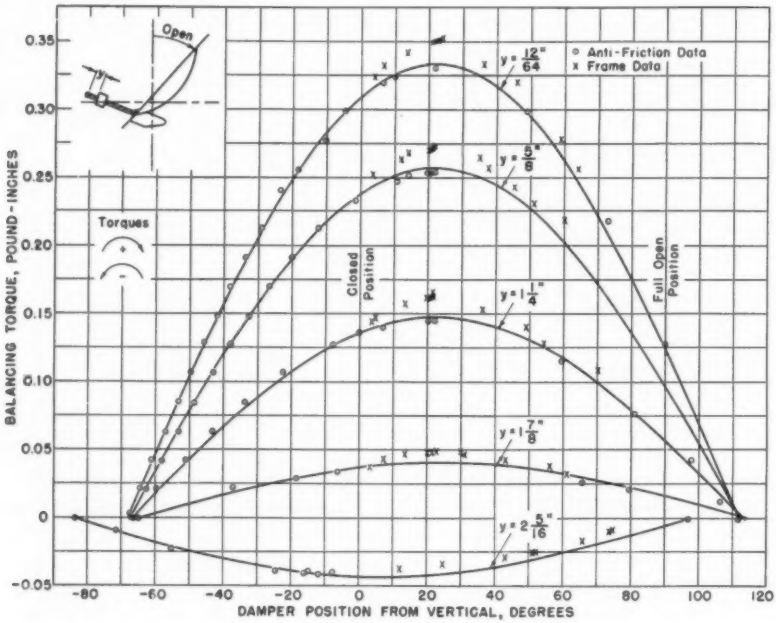


FIG. 5. TORQUE-POSITION DATA FOR DAMPER A

Every effort was made to eliminate friction from the data reported. A special friction study in the case of Damper A gave the comparison shown in Fig. 5. The *anti-friction points* were obtained with the damper axis rolling on drill-rod supports, whereas the *frame* data were taken with the regular sheet-metal supports. General magnitudes of the friction torques are summarized in Table 2. Pertinent interpretations for the individual cases follow.

Fig. 5, Damper A: The similar curves for the same free-rest positions indicate a balance-weight adjustment causing the center of gravity to move along a line passing through the axis of rotation. In this condition, one balance-weight setting would give zero torque for all positions, which would be a case of neutral equilibrium except for friction's restraint.

Fig. 6, Damper B: The joining of all curves in essentially a single point at the full-open position is a consequence of the balance weight being adjusted along a rod suspended below the damper and perpendicular to its face.

Fig. 7, Damper C: The combination of a spring torque with the weight-displacement torque is one means of obtaining a departure from the sine-curve form.

Fig. 8, Damper D: These curves are sine curves with maxima outside the range of normal operation. (The curves, as drawn, refer to the damper mounted in the position sketched. The damper also can be operated if turned upside down with the balance weight range shifted to correspond.)

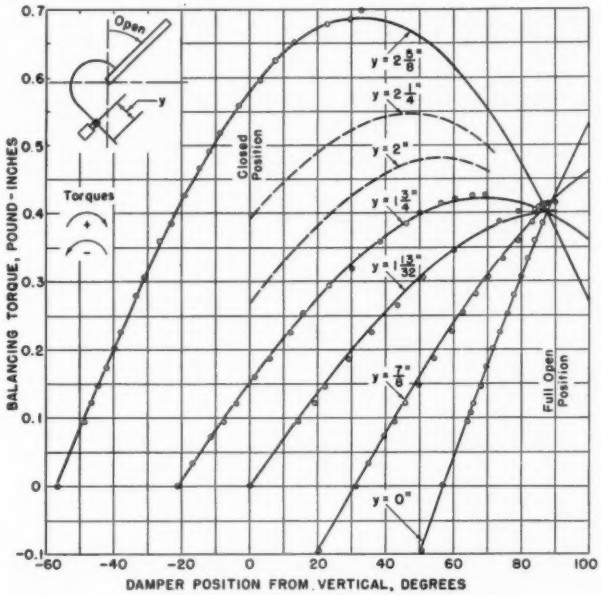


FIG. 6. TORQUE-POSITION DATA FOR DAMPER B

Stability: The equilibrium between the balancing and the weight torques will be stable when the center of gravity is in quadrants III or IV. A change in the balancing torque will merely cause a shift to a new balance position.

TABLE 2—APPROXIMATE SUPPORT-FRICTION TORQUES FOR THE TEST DAMPERS
(Applicable to the Middle Range of the Torque Curve)

DAMPER	FRICTIONAL TORQUE PERCENT OF BALANCING TORQUE
A[a]	1 to 3
B	1 to 2
C	nil
D	2 to 3

[a] On sheet-metal track.

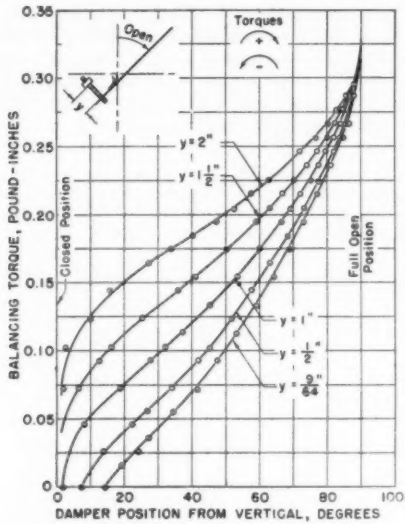


FIG. 7. TORQUE-POSITION DATA FOR DAMPER C

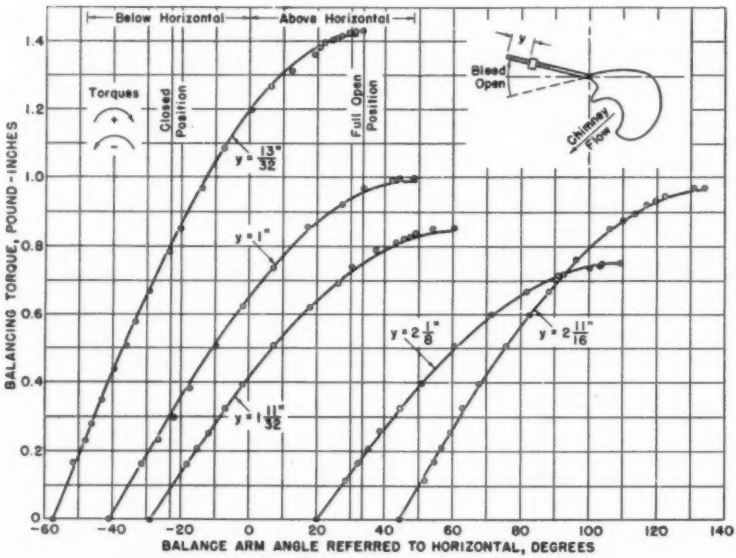


FIG. 8. TORQUE-POSITION DATA FOR DAMPER D

When the center of gravity is in quadrants I or II, the equilibrium may be unstable. Imagine, for example, the damper at rest in some location such as A_4 in Fig 4 and consider the two cases:

1. The balancing torque is decreased by a small amount and held constant at a new value. The weight torque then will exceed the balancing torque, and the center of gravity will fall spontaneously until a new equilibrium is reached in quadrant III.

2. The balancing torque is increased by a small amount and held constant at a new value. The weight torque then will be less than the balancing torque and the damper will swing farther open until it strikes a limiting stop or until the center of gravity finds a new balance in quadrants III or IV.

Design may prevent unstable performance. In some instances, air flow effects may reduce tendencies toward instability, although this has not been examined in the studies thus far.

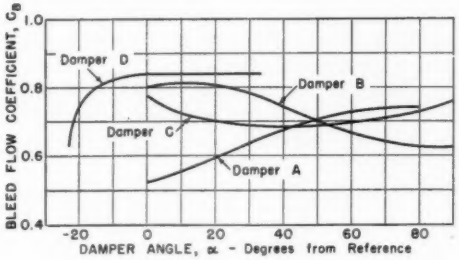


FIG. 9. BLEED FLOW COEFFICIENTS

Design Possibilities: From an appraisal of existing designs, it appears that full advantage has not been taken of an analytical approach to the principles of damper mechanics. Further discussion is given in Appendix A.

DAMPER THROUGH-FLOW CHARACTERISTICS

This phase of the program considers the practical analysis of: (1) the rate of air flow through a damper opening, and (2) the torque exerted on a damper by the pressure difference across it and the flow passing through it. Referring to Fig. 2, these studies culminate in data on two coefficients, one applying to the rate of flow and one to the actuating forces. Damper positions and flow rates, in turn, may be predicted from these coefficients in combination with torque-position analyses.

Coefficient Data: Dampers were mounted on the open end of a straight duct and two types of tests were run: (1) damper angle fixed and air flow rate variable, and (2) damper swinging free and air flow rate variable. Damper angles were measured optically and damper draft was measured at a static pressure tap close behind the damper frame.

For Damper D, involving a combination of bleeding and appliance-flow throttling, two test series were needed: (1) with the appliance-flow duct open

and the bleed entirely blocked, and (2) with the appliance-flow duct sealed and the bleed operating alone.

The *bleed flow coefficient,* employed to represent the flow characteristics of a damper bleed opening, is defined as:

$$C_B = \frac{28.8W}{A_o\sqrt{\Delta P_D\,\rho/\rho_o}} \quad \ldots\ldots\ldots\ldots \quad (2)$$

where

C_B = bleed flow coefficient, dimensionless.
W = flow rate, pounds per second.
A_o = bleed opening area, square inches.
ΔP_D = damper draft, inches of water.
ρ/ρ_o = ratio of intake air density to standard density, dimensionless, where ρ_o is 0.075 pounds per cubic foot.

With intake from the room, there is no velocity-of-approach factor.

Area A_o was taken (by graphical determination) as the minimum cross-sectional area open for flow around the damper. The relationship between A_o and damper angle is controllable by design.

Fig. 9 shows magnitudes of C_B; these were found to be independent of W for $(\Delta P_D\,\rho/\rho_o)$ from 0.005 in. water to the upper test limit of 0.15 in. water. Fig. 10 shows corresponding A_o magnitudes. The reference damper angle here is the same as for the torque data.

For Damper D, tests with the bleed sealed disclosed an *appliance-flow coefficient* defined as:

$$C_A = \frac{28.8W}{A_A\sqrt{\Delta P_B\,\rho/\rho_o}}\sqrt{1-(A_A/A_D)^2} \quad \ldots\ldots\ldots \quad (3)$$

where

C_A = appliance flow coefficient, dimensionless.
A_A = appliance flow opening area at damper, square inches.
A_D = cross-section area of appliance duct approaching the damper, square inches.
ΔP_B = pressure difference across the damper blade, inches of water.
ρ/ρ_o = ratio of air density in duct approaching damper to standard density, dimensionless, where ρ_o = 0.075 pounds per cubic foot.

Equation 3 contains a velocity-of-approach factor. Fig. 11 shows magnitudes of C_A and A_A for Damper D.

A *torque-force coefficient* serves to represent the force acting on a damper at any particular angular position and is defined as:

$$C_F = \frac{27.7\,L}{A_s\,R\cos\alpha}\left(\frac{1}{\Delta P_D}\right) \quad \ldots\ldots\ldots\ldots \quad (4)$$

where

C_F = torque-force coefficient, dimensionless.
L = damper equilibrium torque, pound-inches.
A_s = damper area exposed to pressure difference represented by ΔP_D, square inches.
R = distance between center of pressure on A_s and axis of rotation, inches.
α = damper angle, measured as in Figs. 9 and 10, angular degrees.
ΔP_D = damper draft, inches of water.

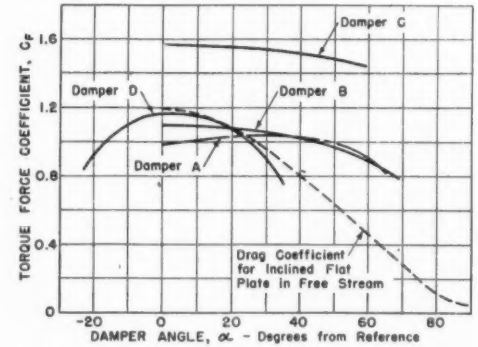


FIG. 12. TORQUE-FORCE COEFFICIENTS

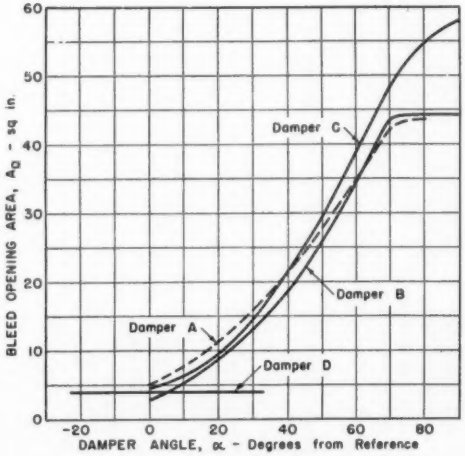


FIG. 10. BLEED OPENING AREAS

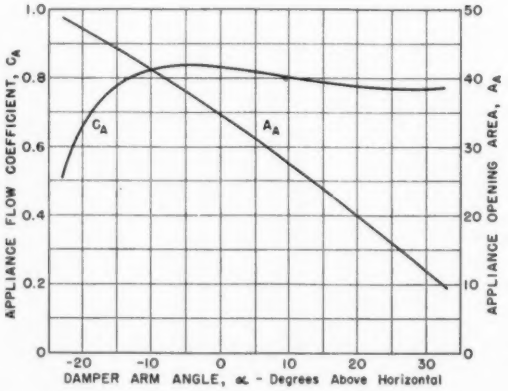


FIG. 11. CHARACTERISTICS OF APPLIANCE FLOW THROTTLING FOR DAMPER D

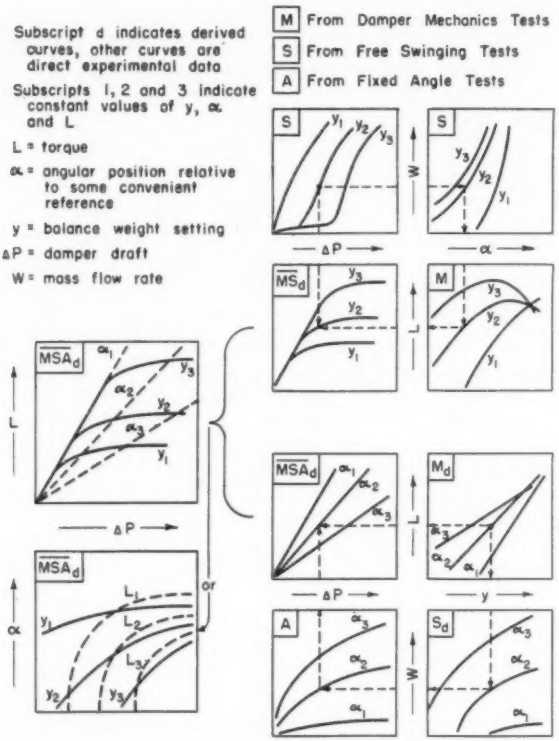

Subscript d indicates derived curves, other curves are direct experimental data
Subscripts 1, 2 and 3 indicate constant values of y, α and L
L = torque
α = angular position relative to some convenient reference
y = balance weight setting
ΔP = damper draft
W = mass flow rate
M From Damper Mechanics Tests
S From Free Swinging Tests
A From Fixed Angle Tests
or

FIG. 13. SCHEMATIC REPRESENTATION OF THE INTER-RELATIONSHIPS OF PERFORMANCE VARIABLES

The product $A_s \cos \alpha$ represents the pressure-surface area projected on a plane that is normal to the duct axis for Dampers A, B and C. For Damper D, A was the area of the curved surface (exposed to ΔP_D) projected on the chord of the surface. The distance R was obtained by assuming that all parts of A_s were exposed to the same pressure-difference force. Table 3 gives magnitudes of A_s and R.

Torque-force coefficients are summarized in Fig. 12. Comparative drag-coefficient data are shown for a flat plate in a free stream[2]. These indicate the

TABLE 3—MAGNITUDES OF A_s AND R

DAMPER	A	B	C	D
A_s, sq in.	51.1	52.1	61.1	36.6
R, in.	1.50	3.53	0.75	4.88

magnitude of the torque-force coefficient which would prevail for a simple flat plate in a free stream with no influence of a supporting frame or adjacent duct walls. That the coefficients for the free plate and the damper should be the same order of magnitude strengthens the principle adopted.

The greater coefficient for Damper C as compared to A and B reflects the influence of sharp edges and a different flow pattern through the bottom of the damper opening.

Coefficients were found independent of W over the test range of drafts.

DISCUSSION

Coefficient data cannot be generalized on the basis of the four exploratory tests reported. The flow-coefficients however are compatible with orifice and nozzle coefficients, and the torque-force coefficients are reasonable in comparison to aerodynamic drag data. Tests to establish coefficients are quite simple, and the present results should serve as a guide for preliminary estimates.

Fig. 13 shows the inter-relationships of Fig. 2 in more specific detail. These curve families are simple and rapid to use, and, from them, it is possible to: (1) predict through-flow performance under any desired condition, and (2) analyze the effects of design changes upon performance. The procedure is based upon physical principles and involves a minimum of experimental data.

Extension of the program as outlined would establish additional relationships to link with those of Fig. 13, until eventually the entire draft system would be studied.

CONCLUSIONS

1. The entire torque *vs.* position relationship is controllable by design, and attainable possibilities do not appear to have been exhaustively analyzed in current practice.
2. Flow coefficients and torque-force coefficients offer a simple basis for treating through-flow characteristics.
3. Results obtained are favorable to the concept that barometric draft controls and draft systems may be designed or matched to meet preestablished requirements in terms of: (*a*) desired appliance performance, (*b*) known chimney performance, (*c*) adequate flow and thermal data for the flue sections connecting the appliance and the chimney, and (*d*) analytical methods of developing or diagnosing draft-control performance.

REFERENCES

1. Unpublished preliminary material on file at A.S.H.V.E. Research Laboratory.
2. *Applied Aerodynamics*, by L. Bairstow (Longmans, Green and Co., London, First Edition, 1920, Page 126).

APPENDIX

Design Considerations and Further Possibilities

THE EQUATION OF THE TORQUE-POSITION CURVE

For rigid rotating damper assemblies, the torque-position relation is Equation 1

$$L = Wr \sin \phi$$

If there are forces involved from springs, bellows, eccentric-suspended weights, or the like, the relationship becomes

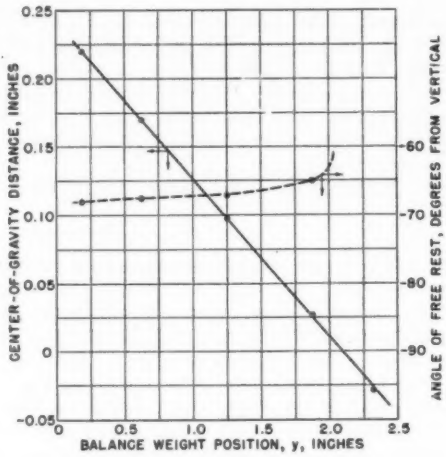


FIG. A. ANGLES OF FREE REST AND CENTER-OF-GRAVITY DISTANCES FOR DAMPER A

$$L = Wr \sin \phi + f(\phi) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where $f(\phi)$ is the net contribution of these latter. Damper C is an example of this type.

Considering only the simple rotating Dampers, A, B and D, the torque principle is well illustrated by giving the center-of-gravity location as a function of balance-weight position; see Figs. A, B, and C.

The center-of-gravity location requires two coordinates, one a distance from the axis of rotation, r, and the other an angle. The angular positions for the torque curves, Figs. 5, 6, 7 and 8, were referred to arbitrary convenient damper positions.

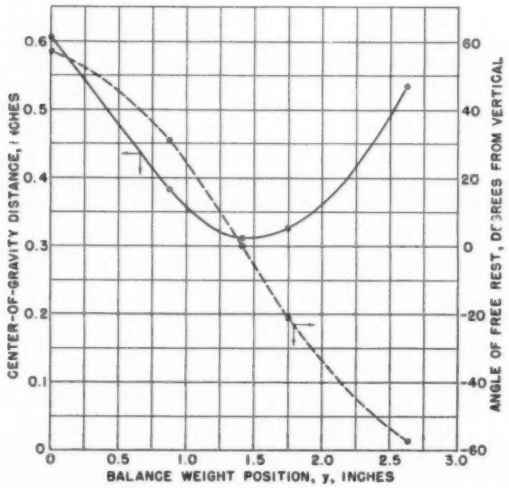


FIG. B. ANGLES OF FREE REST AND CENTER-OF-GRAVITY DISTANCES FOR DAMPER B

Let α denote this angular scale, taking positive α in the direction of an opening damper. Let α_o denote the free-rest position on the α scale, as given in Figs. A, B, and C. Then, with ϕ the angle from free rest,

$$\phi = \alpha - \alpha_o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Equation 1 on the α scale is then

$$L = Wr \sin (\alpha - \alpha_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The method for determing r was to find the sine curve, Equation 7, which gave the best fit to the test points over the entire range of the data.

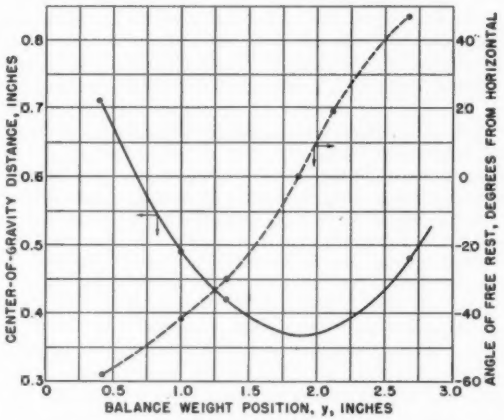


FIG. C. ANGLES OF FREE REST AND CENTER-OF-GRAVITY DISTANCES FOR DAMPER D

DESIGN COMMENTS

In accord with the impression from existing practice that an analytical design approach has not generally been followed, a few comments may be pertinent.

Features of Shape and Assembly: Damper shapes observed have had the simplest form consistent with a good appearance and economy of manufacture from sheet stock. What few cast or machined parts might be employed were of an inexpensive nature. Liberal tolerances were generally evident. An occasional factory-set compensation of the balance system was noted. Parts were generally, but not exclusively, enameled or otherwise rust-proofed. Design Recommendations:

1. When flow-test data are available, it may become justifiable, and equally economical, to consider other than simple circular or rectangular damper shapes.

2. If discrepancies in torque characteristics between supposedly identical dampers are found serious, some sort of factory-set compensation may be justifiable to maintain the design torque relationship.

3. It would be helpful to keep the supports for the rotating parts and all movable elements of the balance-weight adjustment accessible and in view. Disassembly for for cleaning should be simple.

4. Materials should not permit easy corrosion and should withstand temperatures encountered.

Nature of the Balance-Weight Adjustment: Several types of adjustment have been observed. No extensive recognition seems to have been given to the role of this adjustment in locating the center of gravity. Design Recommendations:

1. Give careful study to the influence of the balance weight and its adjustment upon the torque-position curves. A simple relationship between balance-weight position and actuating draft would be very helpful for adjustments in the field, and a setting scale might be developed.

2. Consider how sensitive the location of the center of gravity should be made to slight variations in balance-weight settings. This is largely controlled by the shape of the adjustment member.

Weight of Rotating Parts: Table 1 includes the weight data for the dampers tested. These will serve as representative figures. The less the weight, the less will be the inertial resistance to draft disturbances and the more rapid will be the response of *on-off* appliance operation. Design comments are:

1. A lightweight damper can be made more sensitive than a heavy damper to slight changes in draft conditions; on the other hand, its low inertia may give *flutter* tendencies.

2. A damper made of thicker stock should have greater durability and better withstand thermal expansion effects.

Suspension of Moving Parts: Three types of suspensions have been noted, namely: (1) bearings, (2) axis rolling along track, and (3) springs. What happens to different suspensions when subjected to the dirt and temperatures encountered in the field has not yet been observed, but it seems that a thoughtful design could avoid difficulties. Further design considerations may be justified to obtain different torque curves by means of:

1. A practical suspension which will vary the distance between the center of gravity and the axis of rotation as a function of damper position.

2. Incorporation of springs, bellows, or eccentric weights in the suspension.

Support Friction: Friction at the supports has been found very small for *new* dampers. The effects of severe service conditions have not been investigated. Design refinements are possible to reduce friction still further if needed and economically warranted.

Torque-Position Relation: The relation between balancing torque and angular position is a sine curve for rigid dampers rotating about a *line* axis and which do not employ external forces to influence the balance. Various further possibilities are:

1. Some means of varying the location of the center of gravity of the rotating parts as a function of damper position.

2. Introduction of spring forces, or something similar such as bellows.

3. Use of auxiliary weights to give a torque independently variable with angular damper position.

Designers should give thought to the question of stability as related to the torque-position curve. Automatic adjustment of the torque-position relationship in accord with other combustion controls also may offer advantages.

Center-of-Gravity Location: A design pre-determination of the location of the center of gravity allows the entire torque-curve relationship to be predicted without need for tests. Variations appear innumerable, and exhaustive explorations are not evident.

Flow and Torque Coefficients: Possibilities from changes in damper and flow-passage forms do not appear to have been investigated.

DISCUSSION

R. K. THULMAN, Washington, D. C. (WRITTEN): The work which the authors have concluded in connection with the investigations of barometric dampers is most

significant in underlining the insufficiency of existing knowledge of the behavior of draft systems. The importance of more complete data on draft systems is apparently not fully realized by the profession or by the industry. This is especially the case with those of us who are directly concerned with residential heating.

In the operation of every household fuel-burning device, the draft system requires far more serious consideration than it has received to date. The barometric damper, in its several forms, has been used primarily as a device to limit draft—to keep draft from getting too high. Such use assumes that the draft produced by the flue is more than enough to assure a proper operation of the fuel-burning appliance. For this reason, the operation of draft control devices leaves much to be desired during the critical period of heating-up which follows the initial start of an automatically controlled appliance. The need for a barometric damper which will remain closed, and tightly closed, until the desired limit is reached, is apparent. It is equally apparent that the design-concept of barometric dampers commonly used might be critically examined. Such a critical examination can be profitable to the profession and the industry if the work described in the paper is intelligently applied.

I have observed with great misgiving the tendency of installing contractors to omit draft regulators from residential heating installations. This occurs frequently in installations in houses where the chimneys are low and the available drafts are of the order of one or two hundredths of an inch. An impressive proportion of houses of this type are of limited volume and of tight construction. Few people have realized that an assured supply of air to the fuel-burning appliance is just as important as provision for the disposal of combustion products through a flue. It is an essential part of what the authors term *the draft system.*

It will be unfortunate if the work done so far is not continued for lack of appreciation of its importance to house heating. The need for a comprehensive study of the entire draft system is certainly far more important than many of the research projects now underway. In the Director's letter transmitting a copy of the paper to me for comment, I was invited to suggest *whether any further work should be done in this field by the Society.* It seems to me that further work, expedited by an enlightened appreciation on the part of the Committee on Research, is necessary and essential if the Society's contribution to the public is to be advanced.

I suggest we concentrate on those things which, in my opinion, the public, the industry, and the profession are more directly and more widely concerned. I feel this is more vital than study of heat emission from ceiling panels.

P. R. ACHENBACH, Washington, D. C. (WRITTEN): The authors have very ably illustrated the basic mechanics and flow principles involved in the design of barometric dampers by means of the four specimens tested in the laboratory. Upon first reading this paper I was inclined to criticize it because the information presented and conclusions reached are of little help to the user of barometric dampers or the heating engineer. However the laboratory work completed thus far was only intended to the first two parts of a broader program having at least six objectives.

It should be pointed out that the job of obtaining the proper draft controls will not have been completed when the manufacturer has learned to build a device with the proper torque and flow characteristics. A barometric damper limits the draft by admitting cold air into the system near the thimble thus lowering the average chimney gas temperature and increasing the friction loss in the chimney. Heating devices may be connected to chimneys that are too tall, too large in area, or both, or the chimney may accidentally be of about the right proportions. We do not know the proper size of chimneys to serve heating plants of selected capacities at the present time with any degree of precision.

In my opinion, the laboratory might well complete the third step of the present program involving Damper Tee-Flow Studies, but the barometric damper investigation should not be burdened with the unknowns in the chimney field. I believe the

Research Committee has given a favorable recommendation to the study of draft problems in short chimneys, which has no connection with the study of dampers. A much broader program on chimneys could easily be developed that would include an investigation of the capacity of chimneys for carrying flue gases. Once the proper size chimney for a given structure or a given heating plant was determined, it would be much easier to select the right barometric damper for the job.

HUGH KELLY, Milwaukee, Wis.* (WRITTEN): The paper shows every evidence of careful and painstaking analysis and is of great interest to us as far as the points covered are concerned. The following comments are offered only in the most cooperative spirit. (a) It would have been helpful had actual diagrams of the physical test set up been included. (b) It is stated that coefficient of bleed flow and torque force are dimensionless but it would seem from their use in equations that they have some physical interpretation which could be more clearly brought out. (c) A fuller discussion of the effect of variations in the various coefficients with relation to damper performance would have been very helpful. (d) One phase of the study which is of great importance, which has not been included in this paper, is the relationship of controlled draft to available draft under conditions of varying resistance such as are encountered in actual installations.

It is unfortunate that the complete program as outlined by the T.A.C. on Combustion cannot be completed at this time, as the information so carefully compiled in this paper loses much of its practical value for lack of supporting data which would have been available had the other steps outlined in the program been carried through.

Had the limited scope of this investigation been foreseen it would probably have been of greater value to the members of the Society as a whole to have concentrated on items four and five of the program so that the best possible use could be made of the controls now on the market.

We believe that this program should be carried through to completion at the earliest possible date with the ultimate end in view of setting standards as outlined in item six of the program.

I would like to congratulate the authors who so ably prepared this paper and to express the hope that they can continue the project in the near future.

AUTHORS' CLOSURE (H. B. Nottage): The authors are highly sympathetic towards the comments received. Space limitations forced the manuscript into a compressed styling, which is not appealing to read. The present report is limited to an informative and suggestive status, pending approval of more advanced and practical phases which are contemplated.

Mr. Thulman's comments on the general insufficiency of existing knowledge in the fundamentals of draft systems are subscribed to by the authors. They also share the opinion that much more needs to be done.

Mr. Kelly's points have to do with material which was deleted in the condensed styling and with parts of the program which were not reached.

Mr. Achenbach has correctly foreseen the complications of chimney factors. The authors retain the opinion that the physical principles of making the best match between the several components of a draft system should be part of the present program. Chimney research *per se* is something else again and belongs in a separate project.

In closing this investigation, it is hoped that the spirit of analytical inquisitiveness will not be lost. Draft systems have been *built* for centuries, and it is time that fundamental physical principles should be applied toward a proper understanding through rational investigations.

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* Chief Engineer, Controls Division, Perfex Corporation.

1429

No. 1429

THE CONTROL OF AIR STREAMS FROM A LONG SLOT†

By ALFRED KOESTEL* AND CHIA-YUNG YOUNG**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Case Institute of Technology, Cleveland, Ohio.

A RESEARCH paper on the flow of isothermal air through end supplied slots was published in the 1948 A.S.H.V.E. TRANSACTIONS[1]. This paper is a continuation of the subject with special emphasis on the theoretical flow equations derived for outlets of this type.

The basic outlet dealt with in this paper was a long slot supplied with air from a tapered duct. Mathematical analysis of the desired air flow was the key to the duct shape. The purpose of the design was to eliminate static regain and thereby produce uniform discharge velocities along the entire length of the slot. This feature would prevent lengthwise stream contraction[1] caused by the nonuniformity of air discharge obtained from a slot in the side of a constant cross-sectional duct. It would also facilitate the design and construction of guide vanes for the purpose of obtaining a stream discharge perpendicular to the axis of the slot. It is believed that the basic device tested in this investigation may be the criterion of good slotted outlet design from the standpoint of air distribution.

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† Portions of this paper form part of two theses submitted by the authors to Case Institute of Technology. See References 2 and 3.

* Instructor in Mechanical Engineering, Case Institute of Technology. Junior Member of A.S.H.V.E.

** Graduate Assistant in Mechanical Engineering, Case Institute of Technology.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

THEORETICAL ANALYSIS

Let the following symbols represent the quantities involved, and Fig. 1 the diagram to which the analysis is applied:

X = distance along duct axis to point in question, feet.
V_x = duct axial velocity at distance X, feet per second.
V_s = slot outlet velocity, feet per second.
h_x = duct static head at the distance X, feet.
h_t = duct total head, feet.
h_v = duct velocity head, feet.
θ = emanating air stream angle from slot, degrees.
A_s = slot area, square feet.
A_d = duct cross-sectional area at up-stream end, square feet.
A_x = duct cross-sectional area at X, square feet.
L = slot length, feet.
C_d = coefficient of discharge of slotted orifice.
Q = air quantity, cubic feet per second.
Q_t = total air quantity supplied at duct entrance, cubic feet per second.
S = slot width, feet.

Assuming an ideal fluid, the velocity and static pressure are uniform across the duct cross-section and the total head at any point, X, is constant. Bernoulli's theorem, therefore, states that:

$$\frac{V_x^2}{2g} + h_x = constant = \frac{V_s^2}{2g} \quad \ldots\ldots\ldots\ldots \quad (1)$$

The quantity of flow through a differential length of slot is:

$$dQ = S\,dX\,C_d\sqrt{2gh_x} \quad \ldots\ldots\ldots\ldots \quad (2)$$

where,

$$SdX = dA_s$$

Since the fluid is assumed incompressible, the volume of air entering the supply duct minus the volume discharged from the slot along X must be equal to the quantity flowing within the duct at X. The continuity equation is, therefore:

$$Q_t - \int_0^x dQ = A_x V_x \quad \ldots\ldots\ldots\ldots \quad (3)$$

The foregoing three fundamental equations can be applied to any flow condition in which the air stream approach within a duct is parallel to the outlet. These equations have been successfully applied to ducts of uniform cross-section in which static regain produces a variable flow throughout the slotted orifice. Due to limitation on the extent of the present subject, this topic will be dealt with in a later paper.

The angular direction of the discharged air stream with respect to the slot axis, θ, is related to a vectorial sum of two velocities as shown in Fig. 2. If there is a velocity V_x in the X direction, and a velocity perpendicular to the slot face due to the conversion of the duct static pressure through the outlet,

the resultant will be the slot discharge velocity, V_s. This relationship can be expressed as,

$$tan\ \theta = \sqrt{2\ gh_x}/V_x$$

or, $V_s^2 = 2gh_x + V_x^2$ for the right triangle of Fig. 2, which is also the Bernoulli's equation. (Equation 1).

In order that the discharge angle θ be uniform along the slot, the duct axial velocity V_x must remain constant with respect to X. Therefore, let the duct cross-sectional area A_x be a variable and V_x a constant in Equation 3. Differentiating the equation,

$$-dQ = V_x dA_x$$

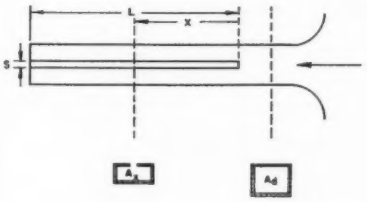

FIG. 1. LONG SLOT WITH AIR SUPPLY FROM ONE END

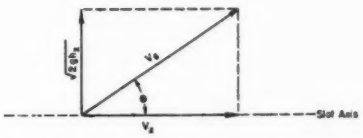

FIG. 2. VELOCITY VECTOR DIAGRAM

(since Q_t is a constant, dQ_t becomes zero) and substituting for dQ in Equation 2:

$$-\ V_x\, d\, A_x = S\, dX\, C_d \sqrt{2\ gh_x}$$

and

$$\frac{-\ V_x\, d\, A_x}{\sqrt{2\ gh_x}} = S\, dX\ C_d$$

If V_x is a constant, according to Equation 1 h_x must also be a constant. Substituting the vectorial relationship

$$-\ cot\ \theta = -\ V_x / \sqrt{2\ gh_x}$$

into the foregoing equation,

$$-\ cot\ \theta\, dA_x = S\, dX\ C_d$$

and integrating between limits,

$$-\ cot\ \theta \int_{A_x = A_d}^{A_x = A_x} dA_x = S\ C_d \int_{X=0}^{X=X} dX - cot\ \theta\ [A_x - A_d] = S\ C_d\ X$$

or

$$A_x = [-\ S\ C_d\ tan\ \theta]\ X + A_d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

which is a straight line equation in the form $y = mx + b$ where m is the slope ($m = -S\ C_d\ tan\ \theta$) and b the y-intercept ($b = A_d$), as shown in Fig. 3. This

equation indicates that the supply duct cross-sectional area, A_x must be linearly decreased under the slot in order to eliminate static regain and to maintain a constant direction of air flow (see Fig. 4).

In Equation 4, when $A_x = 0$, X will be equal to L, the length of the slot. Therefore:

$$-SLC_d \tan \theta + Ad = 0$$

and SL = slot area, A_s
therefore,

$$\cot \theta = \frac{A_s C_d}{A_d} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

The curve of Equation 5 is shown in Fig. 5. The ratio of the velocity head to the total head in the duct may be found by appropriate substitutions. This

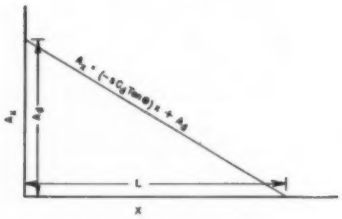

FIG. 3. RELATIONSHIP BETWEEN DUCT SIZE AND SLOT LENGTH

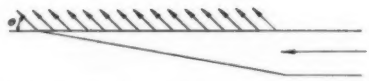

FIG. 4. UNIFORM AIR FLOW FROM A SLOT SUPPLIED BY A TAPERED DUCT

parameter can be applied in selecting the proper supply fan characteristics for this type of outlet.

Substituting the vectorial velocity relationship into Equation 5,

$$\sqrt{\frac{h_v/h_t}{1 - h_v/h_t}} = \frac{A_s C_d}{A_d} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

The curve of Equation 6 is shown in Fig. 6.

COMPARISON WITH TEST

In order to verify Equations 5 and 6 a 10-ft sharp-edged slot in the side of a tapered supply duct 8.06 × 8.06 inches square at the upstream end was constructed. The width of the slot was adjustable so that the area ratio, $A_s C_d / A_d$ could be varied. A coefficient of discharge for the sharp-edged slot was assumed to be 0.61. The maximum slot width investigated was 3.773 in., ($A_s C_d / A_d = 4.25$), and the minimum width was 0.25 in. ($A_s C_d / A_d = 0.282$).

Pressure, flow, and air discharge angle measurements were made as described in the first paper[1]. Air supply to the test duct was from a large plenum chamber through a rounded entrance nozzle. The air quantity was measured by calibrated orifices and corrected for system leakage. Sidewall static taps were made to *A.S.M.E.* code specifications, and the pressures were read on calibrated in-

clined gages to 0.002 in. of water. All tests were run under isothermal conditions with air volumes ranging from 140 to 1350 cfm. The entire experimental investigation covered a period from 1947 to 1949.

The theoretical curve of Equation 6 (see Fig. 6), which indicates the relationship between the pressure distribution in the duct and the area ratio, shows good agreement with the test points. Each point represents the average pressure ratio obtained for three or four rates of flow. The slight deviation of the data from the curve between area ratios of 2.26 and 4.25 is due primarily to the difficulty of measuring the extremely low duct static pressure for the volume of air discharged in the higher area ratios (large slot area).

Fig. 5 shows the relationship between the constant discharge angles along the slot length and the area ratio. The final direction of the discharge air stream

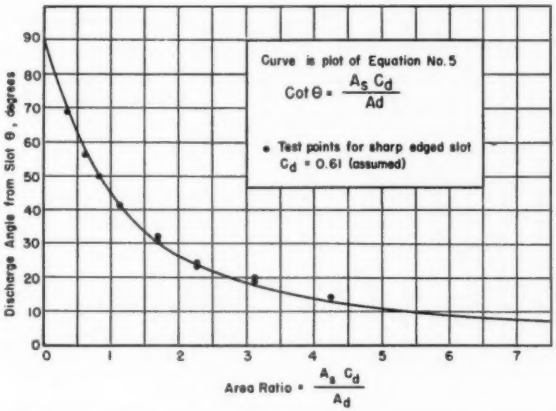

FIG. 5. RELATIONSHIP BETWEEN DISCHARGE ANGLES ALONG THE SLOT AND AREA RATIO

within the space with respect to the slot axis will be the same as the air flow angle Θ. The angles measured by a null-type double impact tube air protractor*[1, 2, 3] show a better agreement with the theoretical curve at the higher area ratios than the pressure ratio curve of Fig. 6. This is due to the fact that the flow angles indicated by the air protractor are a direct measurement of the relationship between the static and velocity pressures. On the other hand, the ratio of the velocity head to the total head involves an orifice flow reading and a static pressure reading at the duct inlet, both of which are subject to errors in reading.

APPLICATION OF TURNING VANES

If it is desired to obtain an air discharge perpendicular to the duct axis, turning vanes or deflectors should be applied. The tapered duct design is best suited

* An indicating thread read on a protractor scale and actuated by the slot air flow was used as a check on this instrument.

for turning vanes due to the fact that the discharge angles remain constant. This feature would make the use of turning vanes along the outside surface of the slotted orifice more practical from the standpoint of design and construction. This would not be possible with a constant cross-sectional duct supplying a slot since each vane would have to be designed to meet the condition of flow at a particular point along the slot which is, of course, a variable due to static regain.

The amount of turning required of a vane would depend on the discharge angle. Deflecting vanes should be located on the outside surface of the slot since the direction of the emanating air stream is determined after the duct static head is converted into velocity through the slotted orifice.

Fig. 7 shows an installation tested for a constant discharge angle of 42 deg. One design factor of prime importance is the ratio of vane depth to vane spacing.

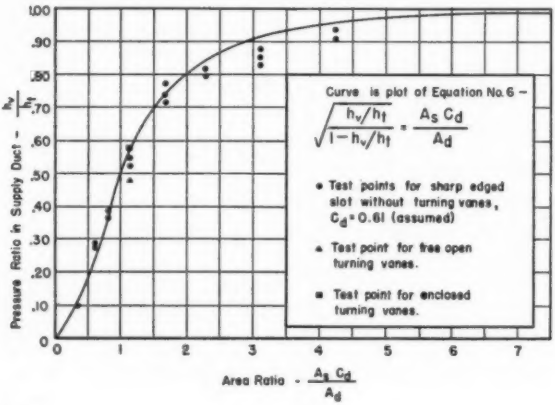

FIG. 6. RELATIONSHIP BETWEEN DUCT PRESSURE RATIO AND AREA RATIO

According to the HEATING, VENTILATING, AIR CONDITIONING GUIDE, vanes should have a depth of one to two times the spacing. If the ratio is less than one, effective control of the discharging air stream cannot be obtained. Little improvement is realized by increasing the ratio beyond two. Based on this, turning vanes were tested for ratios of one and two. The vane depth was 3 in. with a curvature designed for a discharge angle of 42 deg. A smoke pattern showed that a vane ratio of one corrected the discharging air stream from 42 deg to approximately 80 deg to the axis of the duct. A ratio of two turned the stream very close to 90 deg.

For appearance the vanes should be enclosed or *boxed in* as shown in Fig. 7 so that they will not protrude out into the occupied space. The effect of free open and enclosed vanes on pressure ratio (h_v/h_t) is shown in Fig. 6. Any change in the coefficient of discharge from the assumed value of 0.61 due to the vane installation, will deviate the experimental points from the theoretical curve. As a result the actual area ratio will not be the same as the one determined from

the physical dimensions of the slot and duct, and from the coefficient of discharge for a free open, sharp-edged slot.

A turning vane, if designed correctly, should have the shape of an air foil so that separation of the deflected air stream from the vane surface does not take place. This, no doubt, would be impractical and uneconomical in actual air distribution installations. Vanes of single thickness are much more easily fabricated than vanes of variable thickness as in an air foil. In order for a guide vane of single thickness and a given height to be efficient, the curvature should be as flat as possible without any abrupt turns. This can be done if a circular arc is used which is tangent to the discharge angle at one end and tangent to a line

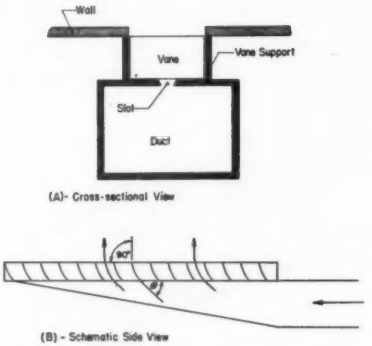

FIG. 7. (A) INSTALLATION OF ENCLOSED VANES ON THE SLOT. (B) APPLICATION OF TURNING VANES TO SLOT

perpendicular to the slot axis at the other end. The radius of the arc would then be determined by the vane height and the discharge angle. This is illustrated in Fig. 7.

Velocity traverses were taken of the air stream leaving the vanes. It was found that the magnitude of the velocities was at a maximum near the face of each vane and at a minimum immediately behind the vane. In the case of the wider spacing (1:1 ratio) the variation in velocities along the slot could easily be detected by passing a hand over the discharge. For the narrower spacing (2:1 ratio) the velocity variation could hardly be perceived except by pitot tube measurement.

PRACTICAL APPLICATION

Standardization of vane design can be devised in order to reduce costs of fabrication and installation. This could readily be done by selecting two or three practical discharge angles to be used in all types of slotted outlets. From the equation

$$\cot \theta = A_s C_d / A_d$$

various slot lengths, widths, and supply duct sizes can then be chosen for the given discharge angles. Sections of bracketed guide vanes already preset for the selected discharge angles could then be installed along the face of the slotted outlet. Fig. 8 shows a section of vanes used in the tests.

The necessary supply fan characteristics and power requirements for the outlet would be determined by the pressure ratio (h_v/h_t) corresponding to the selected discharge angles and the quantity of air to be handled. Duct losses between the outlet and the supply fan should be taken into account so that the pressure ratio can be corrected to the required conditions at the fan.

Care should be taken in assuming the correct coefficient of discharge that is to be used in the area ratio. Factors such as orifice construction (sharp- or round-edged slots) and the method used in enclosing the vanes may affect the discharge coefficient.

Under certain conditions guide vanes may not be practical for a slotted outlet. The discharge angle θ may then be selected on the basis of the desired direction

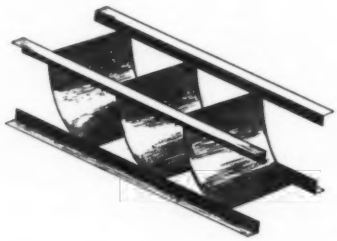

FIG. 8. SECTION OF PRE-SET, BRACKETED TURNING VANES USED IN TEST

of the stream within the conditioned space. Air distribution requirements within the space would then be the governing factor in selecting the proper direction of the supply stream discharge. It should be kept in mind that in choosing too large a discharge angle an impractical design may result, namely, one with a small slot area, a large duct size, and a high static pressure.

EXAMPLE OF PRACTICAL APPLICATION

As an example of the design of a slot-type air distribution unit, assume that it is desired to distribute 1500 cfm of primary air from a slot 20 ft long located in the side of a wall of approximately the same length. The supply air is to be distributed in equal quantities per foot of slot length and in a direction nearly perpendicular to the duct axis. This is necessary to uniformly absorb a heavy cooling load within the space without excessive temperature gradients.

The velocity in the supply duct is not to exceed 2000 fpm and the slot is to be sharp-edged. Throw limitations will not be considered in order to simplify the example.

Required: Duct size, slot width, static pressure in duct, and a means of introducing the primary air uniformly into the space.

Solution: Two possible solutions will be considered for comparison: namely, a design with (1) Turning vanes set for a perpendicular stream discharge installed on the slot face (see Fig. 7), (2) A slot (and duct size) without turning vanes to give an air stream, for example, of 80 deg to the slot face (see Fig. 4). Assume that a standard set of bracketed, 45 deg turning vanes is available for installation. The solution for each design is tabulated in Table 1.

The tabulated results indicate an impractical situation for the slot design without turning vanes. The slot width is exceedingly small and is, therefore, impractical. The required duct pressure is 32 times as great as the pressure for the vaned slot, resulting in excessive fan power requirements and in objectionable slot velocities. Installation of turning vanes would be the practical solution in this case. The vaneless slot design could be somewhat improved by decreasing

TABLE 1—DUCT DESIGN COMPARISON

	80 DEG STREAM WITHOUT VANES	90 DEG STREAM WITH VANES SET FOR 45 DEG
1. Duct size, sq in. $(1500 \times 144)/2000 = 108$ sq in. or	12″ × 9″	12″ × 9″
2. Slot width, inches: Equation 5 Assume $C_d = 0.61$		
Cot 80 deg × $108/(12 \times 20 \times 0.61) = s$	0.13	
Cot 45 deg × $108/(12 \times 20 \times 0.61) = s$		0.74
3. Duct velocity pressure corresponding to 2000 fpm, in. water	0.25	0.25
4. Duct pressure ratio h_v/h_t: Equation 6 (or from Fig. 6)	0.03	0.50
5. Duct static pressure, in. water, (assume turning and frictional losses negligible) Item 3/Item 4 − Item 3	8.10	0.25

the allowable discharge angle, say to 70 deg. This would be permissible if a satisfactory air distribution pattern could be maintained. Furthermore, a decrease in angular direction may cause impingement or *splashing* of the primary air on the adjacent wall at the downstream end of the slot. Another possible improvement in design would result if a larger duct size were selected. If the same discharge angle and area ratio were used, the slot width could then be increased resulting in lower duct pressures.

ACKNOWLEDGMENTS

The authors are indebted to Dr. S. D. Black of the Aeronautical Engineering Division at Case Institute of Technology for his help and suggestions in the theoretical analysis of this special air flow problem, and to Prof. G. L. Tuve for his guidance in carrying out this investigation. The authors also wish to thank Miss Ruth Ehrenfeld for her preparation of the diagrams.

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2. Air Flow Through Outlets in the Side of a Long Duct, by Alfred Koestel (A thesis submitted to Case Institute of Technology in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering, 1949, 83 pages).

3. Control of Discharge Angle of Air from a Long Slot, by Chia-Yung Young (A thesis submitted to Case Institute of Technology in partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering, May, 1949, 83 pages).

DISCUSSION

G. B. PRIESTER, Baltimore Md. (WRITTEN): The authors are to be complimented on their presentation of the theory, correlating test data, and application information on air streams from a long slot.

As previously pointed out, however, I do not believe that the friction loss in the tapering duct should be completely ignored even though it may be a small item. The friction drop for 1500 cfm at 2000 fpm, as used in the example, is practically ½ in. of water per 100 ft of duct.

Since in practice the desired throw usually is fixed, it would be helpful if a practical example could be presented in which the throw is, say 30 ft. It is also suggested that an example using a lower duct velocity, say 1000 or 1500 fpm, be included.

C. M. WILSON, New York, N. Y.: The timeliness of this paper is appreciated, as we have been conducting a lot of research on straight line diffusers. One big problem has been the approach to the diffuser. I believe this theoretical approach and test data are going to be very helpful, particularly to determine air flow patterns before the air hits the slot. Our problem has been to deliver air evenly through a straight line diffuser as this unit is preferred by many architects.

We conducted research last winter and tried to design a number of ceilings which were acoustically correct and combined good diffusion, color and light, and an attractive appearance. We were surprised to learn that the straight line diffuser or the slot, appearance-wise, fitted a number of good designs better than many other type outlets now on the market.

While we have never been able to make a simple slot perform well in a test room regardless of the approach or even distribution, we nevertheless feel that this work should be continued. We have had to break up the air supply into as many as five streams to obtain good mixing. Before this breaking up of the air stream, the air came out of a slot, particularly when a high temperature difference was present, and fell on a room occupant's head.

I hope this research will determine not only proper methods of delivering air from a slot, but also how best to control it after leaving the slot.

CYRIL TASKER, Cleveland, Ohio: The paper presented today is a report on one portion of a long range and comprehensive research program on air distribution which has been carried on for a long time under the Society's Technical Advisory Committee on Aid Distribution of which Mr. Szekely was Chairman for several years.

The overall program on air distribution is made up of a number of important parts. One of these, fundamental studies on ventilation jets, is rapidly nearing completion at the Society's Research Laboratory, and we expect to present to the membership in the next 18 months a number of papers from these studies.

The work at Case Institute, of which this paper represents only a small part, has been underway for several years, has covered all types of outlets and has produced a whole series of valuable papers.

Studies at Kansas State College under Professor Linn Helander are concerned with the downblow of heated and cooled air streams. One valuable paper has already been presented and more are to come.

Under Dean L. G. Miller at Michigan State College studies are being made of the friction in take-offs in rectangular ducts.

Society studies on air distribution are extensive, but the subject is an extensive and complicated one. Slowly but surely we are compiling a great deal of valuable information; this paper is an excellent example of how worthwhile results can be obtained by distributing our problems among colleges interested in assisting the Society in its research program.

Sometimes we are apt to overlook the changes and additions that have been made to THE GUIDE as a result of Society work on air flow. It takes a long time to get some of the results translated into practice but this is being accomplished.

This paper is another example of the way in which information of practical value is being secured for the benefit of the membership as a result of the Research program.

P. R. ACHENBACH, Washington, D. C.: This work at Case Institute of Technology is very timely in connection with warm air perimeter heating. Where large picture windows are used it is desirable to deliver heat near the base of the windows to counteract the cold currents created by them. The use of a long slot below large windows extending the full length of the window in connection with perimeter warm air heating appears to be a very good way to provide adequate heating where it is most needed in such cases. This paper which shows how the warm air can be discharged in the proper direction from a long slot will be very helpful in designing such installations.

LESTER T. AVERY, Cleveland, Ohio: Many men attending these meetings feel that the papers presented here are too theoretical. I suggest to them that the greatest problem one has in handling air for air conditioning (cooling, heating, dehumidifying, humidifying) is air distribution, and many people are inclined to take air distribution for granted, from a catalogue, without attempting to understand the problems of air distribution.

I feel that papers of this kind are far from being too theoretical or too far removed from practice. They are years ahead of catalog data from the manufacturer. Unless one takes the time to read these papers and understand them, he will be two to three years behind the art. Through research of this kind we are learning some of the vagaries of handling air.

I think it is a fine paper, and I think this Society is the only place where such information will be developed.

R. J. ABRAMSON, Chicago, Ill.: We have in suburban Chicago (Highland Park), a residence constructed for Mr. and Mrs. Slavin. One section of the house is heated by underfloor ducts supplying air to perimeter outlet slots. Another section of the house has perimeter intake air slots connected to underfloor ducts. This building has been occupied eight months. We are keeping careful records and hope to present actual test results next year for this type of installation.

Winter heating, using perimeter slots and underfloor ducts has been very satisfactory. This building is located on a bluff at the shoreline of Lake Michigan with maximum possible exposure. We have had absolutely no frosting of the large windows while maintaing the design humidity conditions within the space. Vertical supply air velocities at the perimeter at the six foot level were 20 fpm. The concrete floors average 69 F, ambient air temperature at 60 in. is 72 F, air temperature 6 in. below ceiling is 74 F.

L. P. SAUNDERS, Lockport, N. Y.: Professor Tuve requested such information from any of us concerning distribution or the methods of determining distribution of air. Some years ago we were interested in trying to sell to a refrigerator manufacturer small condensers to be mounted on a bracket carrying the motor which drove the compressor and also handled the fan for the condenser. We wanted to mount this

special condenser we developed directly on the shaft axis so that as the motor moved to keep the belt adjustment tight, the condenser moved with it.

We went to the manufacturer to talk about it, and were interested to learn that they were investigating the panel type or plate type of condenser which was mounted on the back of the refrigerator. The research engineer of this firm told me that in their investigation work they had run into a great deal of difficulty in trying to duplicate hot room test results. That is, they would set one refrigerator in a hot room in a particular place, and when these results were analyzed in comparison with previous results, they did not correspond. It was finally concluded that the difficulty was due to the change in thermal conditions of the air actually in the room.

In order to determine the air flow pattern within the test room, it was completely darkened, aluminum dust was sprayed into the air, and through the medium of either flashlight photography, or spotlight photography with high speed machinery, photographic exposures were made. From then on when a comparison was desired between one test and another, the refrigerator was returned to the original position. Perhaps this method of studying air flow patterns may be useful to the authors.

AUTHORS' CLOSURE (G. L. Tuve): I want to thank each of the discussors for his comments. These remarks bring up one thing that indicates a void in our information. How long is a long slot? We don't know. We know that a slot a quarter inch wide and three feet long isn't a long slot; how long it has to be before it becomes a long slot, we don't know. How does a long slot perform compared with a short one?

Mr. Wilson suggested that air from a slot is likely to dump down on people's heads. Air doesn't stay in a ribbon or flat stream after it leaves the outlet. It tends to bunch together into a round stream just as quickly as possible, and of course, if it is cold, the people just beneath the center of the stream will feel a draft. There is a good deal more to be found out about how to treat a slotted outlet with respect to dividing it up. Dividing up a slotted outlet that is five feet long into one-foot sections, spaced six inches apart doesn't do any good to speak of, so the question is how far the slots have to be separated before they are going to form separate streams.

We have found in work on perforated panels that even a panel with five or ten percent free area tends to act like a single stream coming out of a single hole. There are many unanswered questions, as the discussers have indicated. We are glad to learn more about the practical applications.

Mr. Priester's comment was on the friction in the duct. That is, of course, of some importance but much less than would be expected. It seems to us that for most practical problems, the friction in the duct will not affect the result nearly as much as the uncertainty about the stream of air after it gets into the room. We are very anxious to have further information as to what happens to this stream in the room. We have three men spending full time studying air flow in the room during the summer. They have found these studies difficult, as the room must be closed up to get rid of drafts, and it is 100 F outdoors. We invite engineers who have information that they would care to share, to contact us so that we may exchange information.

1430

No. 1430

PRESSURE LOSSES OF TAKE-OFFS FOR EXTENDED-PLENUM DUCT SYSTEMS

By J. W. HOLL*, S. F. GILMAN**, R. J. MARTIN† AND S. KONZO††, URBANA, ILL.

THE EXTENDED-PLENUM type of air conditioning duct system is characterized by a main duct of the same size and shape throughout its length, designated as the *extended plenum*. The take-offs connecting the extended plenum with the branch ducts are one-piece fittings of simple construction.

Because of the overall simplicity of extended-plenum systems, both the fabrication and installation costs are often less than those of conventional trunk duct systems. Although these advantages have been recognized for some time, complete information for design purposes is not available in the literature.

An investigation of the pressure losses of various types of 7-in. diameter take-offs installed one at a time on a 12-in. × 8-in. extended plenum has been reported by other investigators.[1] As a result, certain sizes and shapes for side and top take-offs were accepted by the industry as standard. Since the reported results were restricted to one size of branch duct and one size of plenum, further research with other sizes of branch ducts and plenums is desirable.

OBJECTIVE OF INVESTIGATION

The objective of the investigation reported in this paper was to determine the performance characteristics of extended-plenum and branch-duct combinations, such as are normally used in residential air conditioning applications,

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

utilizing standard take-offs. The items which were investigated in the laboratory and which are discussed in this paper are the following:

1. The performance of standard side and top take-offs when used alone on two different plenums.

2. The effect of the relative locations of two or more take-offs on the pressure loss of individual take-offs.

3. The application of the laboratory results to the design of extended-plenum systems.

DESCRIPTION OF APPARATUS

The principal details of the laboratory apparatus are shown in Fig. 1. The four orifices in the orifice box were used singly or in parallel to measure the total air flow rate in cubic feet per minute, depending upon the volume range desired. The speed of the centrifugal fan was varied over a wide range by the use of

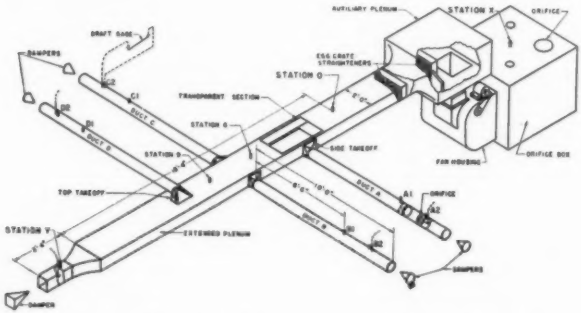


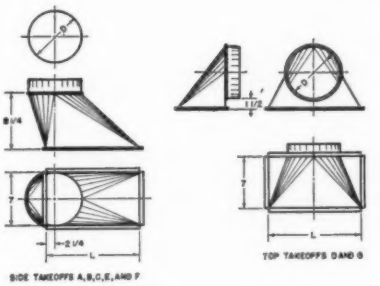
FIG. 1. GENERAL ARRANGEMENT OF APPARATUS

two adjustable rheostats in conjunction with a direct current motor. The fan discharged the air from the orifice box into the auxiliary plenum and thence into the extended plenum. The function of the auxiliary plenum, nozzle, and egg-crate straighteners was to provide a uniform velocity profile at Station O. The air was then discharged from the extended plenum through the branch ducts and plenum exit section. The dampers were adjustable and were used to establish and control the desired conditions. A 3-ft section of the extended plenum adjacent to the first take-off connection was fitted with a transparent top and bottom to permit visual observation of the air flow pattern in the extended plenum and take-off. Qualitative flow patterns were determined by probing this section with a silk thread attached to a small diameter rod.

Studies were conducted with two 24-in. wide extended plenums, one of which was 8-in. deep and the other 12-in. deep. During these studies, one or more branch ducts were connected at different locations on the extended plenum. The top and side take-offs used in this investigation were the types accepted as standard by the *National Warm Air Heating and Air Conditioning Association.* The construction details of these take-offs are shown in Fig. 2; they have been

designated by the letters A through G for ease of reference. These 7-in., 8-in., and 9-in. diameter take-offs connected corresponding sizes of branch ducts to the plenums.

Referring to Fig. 1, Station O was the primary energy measuring station, and any station along the plenum downstream from Station O was designated by its distance, in feet, from Station O. Thus, Station 6 and Station 9 were located 6 ft and 9 ft, respectively, downstream from Station O. The static pressure at Station O was measured by means of a piezometer ring, and the total air flow at Station O was determined by means of the orifice box. The static pressure tap located at the center of the top of the orifice box, Station X, was used to measure the pressure differential across the one or more orifices in use. The static pressure at any station in the plenum downstream from Station O, such as Stations 6 and 9, was measured by a Pitot tube. The volume (cfm) leaving the exit section of the extended plenum was measured at Station Y.



Branch Duct

Take Off Designation	Diameter D
A	7
B	7
C	7
D	7
E	8
F	9
G	8

Take Off Classification	Dimension L
7S	12
7S	12
7S	12
7T	12
8S	14
9S	16
8T	14

FIG. 2. DETAILS OF TAKE-OFFS

The energy measuring stations in the branch ducts were designated by the numeral 1 following their identifying letters, and the flow rate measuring stations were designated by the numeral 2 following their identifying letters. For example, Stations A1 and A2 in Fig. 1 refer to the respective energy and flow rate measuring stations in Branch Duct A.

CALIBRATION OF APPARATUS

A micromanometer[2] was used to calibrate the 16 inclined draft gages to obtain calibration curves for correcting the gage readings to their true values.

The flat-plate orifices in the orifice box were initially calibrated in place by means of a circular duct and Pitot-tube traverses. After calibration, any air flow rate between 80 and 2400 cfm could be determined from the static pressure reading at Station X.

At the end of the extended plenum, a total-pressure tube and a static-pressure tap were used to measure the velocity pressure at the center of the 8-in. × 8-in. duct at Station Y downstream from the converging nozzle. During calibration of this station, all branch ducts were closed, and the relation between the

velocity pressure at Station Y and the volume (cfm) entering the orifice box was obtained. Any air flow rate between 180 and 1800 cfm leaving the plenum could then be determined from the velocity pressure at Station Y.

A total-pressure tube and static-pressure tap similar to that employed at Station Y were used to measure the velocity pressure in each branch duct. During calibration of any branch duct, the other branch ducts and the exit section of the plenum were closed and the relation between the velocity pressure just downstream from the branch duct orifice and the volume entering the orifice box was obtained. Any air flow rate between 80 and about 600 cfm could then be determined from the velocity pressure reading in the branch duct.

General Procedure and Compilation of Data

For a given configuration of the apparatus, the dampers were set at desired positions. The total volume (cfm) entering the apparatus was then varied over a wide range in about four steps, by varying the fan speed, and the various pressure readings were recorded. The dampers were then reset to other positions and the same procedure was repeated. By this method, the volume flowing in the branch duct and in the extended plenum were varied over a wide range. The mean velocity at any cross-section of the apparatus was calculated from the air flow rate and the cross-sectional area.

In order to evaluate the pressure loss of a take-off, it was necessary to obtain pressure readings which were not influenced by the disturbances caused by the air flowing from the plenum and through the take-off to the branch duct. By probing the cross-section with a Pitot tube upstream from the take-off in the plenum and downstream from the take-off in the branch duct, desirable locations for measuring stations were obtained. The plenum energy measuring stations were located from 1½ ft to 3 ft upstream from the take-off, depending upon the number and spacing of the take-offs in use. The branch duct energy measuring stations were all 6½ ft downstream from the take-off.

For the purpose of illustrating the method used to calculate the pressure loss of a take-off, Fig. 3 will be utilized. The total pressure in the plenum was expressed as:

$$(TP)_P = (SP)_P + (VP)_P \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

$(TP)_P$ = total pressure in the extended plenum upstream from the take-off at Station P, inches of water.
$(SP)_P$ = static pressure as measured at Station P, inches of water.
$(VP)_P$ = velocity pressure corresponding to the mean velocity of flow at Station P, inches of water.

The total pressure in the branch, $(TP)_B$, was evaluated in a similar manner. The pressure loss between Stations P and B is

$$H = (TP)_P - (TP)_B \quad \ldots\ldots\ldots\ldots \quad (2)$$

where

H = pressure loss between Stations P and B, inches of water.

The take-off loss was separated from the normal duct friction loss in accordance with usual procedure of determining fitting losses on a ***no-length*** basis.

The take-off loss was therefore defined as

$$L_T = H - L_P - L_B \quad \ldots \quad (3)$$

where

L_T = take-off loss, inches of water.
L_P = normal friction-pressure loss in the plenum between Station P and the centerline of the branch duct, as determined from the A.S.H.V.E. friction chart,[3] inches of water.
L_B = normal friction-pressure loss between the take-off and Station B, as determined from the friction chart,[3] inches of water.

Substituting Equation 2 into Equation 3, there results

$$L_T = (TP)_P - (TP)_B - (L_P + L_B) \quad \ldots \quad (4)$$

This equation was used to compute the take-off loss.

In order to obtain a dimensionless expression for the take-off loss, each value of L_T, in inches of water, obtained from Equation 4 was divided by the velocity pressure (also in inches of water) corresponding to the mean velocity of flow in the branch duct. For the studies conducted with a single take-off installed on a plenum, the resultant expression was designated the *independent loss coefficient*, λ. Thus,

$$\lambda = L_T/(VP)_B \quad \ldots \quad (5)$$

where

$(VP)_B$ = velocity pressure corresponding to the mean velocity of flow in the branch duct, inches of water.

When experiments were conducted with two or more take-offs installed on a plenum, the pressure loss of each take-off could conceivably be affected by the presence of the others. In order to differentiate between the two experimental conditions, the dimensionless expression obtained by the same procedure used to obtain Equation 5 was designated the *interference loss coefficient*, θ. Thus,

$$\theta = L_T/(VP)_B \quad \ldots \quad (6)$$

in which it is understood that in Equation 6, L_T is evaluated from the data obtained with two or more take-offs installed on a plenum.

The difference in the loss coefficients as determined by Equations 5 and 6 was designated the *take-off interference* and served as an index of the interference caused by the proximity of two or more take-offs.

PRELIMINARY ANALYSIS

In an effort to determine the principal variables involved in this investigation of extended plenum systems, an analytical approach to the problem was made. From a review of the literature and from principles of fluid dynamics, it was expected that the independent loss coefficient, λ, would be influenced by the velocity in the plenum just upstream from the take-off, the velocity in the branch duct just downstream from the take-off, the size and shape of the plenum and take-off, and the density and absolute viscosity of the air. Since air at room temperature and at relatively low velocities was to be used, the effects of com-

pressibility and heat transfer could be safely ignored. The general relationship between the variables considered to be of significance can be expressed as

$$\lambda = f(V_B, V_P, D_B, D_P, \rho, \mu) \quad \ldots\ldots\ldots\ldots (7)$$

where

V_B = mean velocity in the branch duct just downstream from the take-off, feet per second.
V_P = mean velocity in the plenum just upstream from the take-off, feet per second.
D_B = characteristic length of the take-off and branch duct, feet. See Fig. 3.
D_P = characteristic length of the extended plenum, feet.
ρ = density of the air, pounds per cubic foot.
μ = absolute viscosity of the air, pounds per foot second.
f = symbol indicating a functional relationship.

The *characteristic lengths* D_B and D_P may warrant explanation. The selection of the particular dimension to represent a characteristic length is arbitrary for any one combination of plenum, take-off, and branch duct such as shown in Fig. 3. This is true because in describing the size and shape of an object, it is sufficient to select any one dimension; all other dimensions can then be expressed in terms of the selected dimension. For example, if the dimensions of a plenum were known, the width could be expressed as some multiple of the depth, D_P.

Applying the methods of dimensional analysis to Equation 7, there results

$$\lambda = \phi\,(V_B/V_P, D_B/D_P, V_B D_B \rho/\mu) \quad \ldots\ldots\ldots\ldots (8)$$

The last term in the right hand member of this equation is the expression for the Reynolds number referred to the branch duct. Therefore, Equation 8 can be expressed as

$$\lambda = \phi\,(V_B/V_P, D_B/D_P, R_B) \quad \ldots\ldots\ldots\ldots (9)$$

Equation 9 states that the independent loss coefficient, λ, is some function of the velocity ratio, V_B/V_P, the length ratio, D_B/D_P, and the Reynolds number in the branch duct, R_B. The equation does not provide information as to the form of the functional relationship that exists between λ and the three ratios; it states only that a general functional relationship exists between the dimensionless groups. However, considerable information can be obtained from an inspection of Equation 9. One important fact is that the dimensionless groups are independent; *i.e.*, any two can be held constant while the third is varied in any desired manner. For example, if a particular plenum, take-off, and branch duct combination is studied, the value of the length ratio D_B/D_P will be constant. If the total flow rate is varied, but the flow rate in the branch duct is held constant by controlling the flow with a damper, R_B will be constant and the velocity ratio, V_B/V_P, will vary. Thus, the effect of each ratio on λ can be determined experimentally by holding the other two constant. Since the research program is to be limited to a relatively narrow range of air velocities and plenum and take-off sizes, it is probable that the effect of at least one ratio will be insignificant and can be ignored. Any ratio is insignificant if it is essentially constant over the range in which the experimentation is conducted.

Certain conclusions can be drawn from a consideration of the length ratio, D_B/D_P, in Equation 9. If the characteristic lengths D_B and D_P of one configuration of the laboratory apparatus were, for example, twice the respective characteristic lengths of another configuration, the ratio D_B/D_P would be the

same for both configurations. For this condition, Equation 9 indicates that λ would depend only on the velocity ratio and Reynolds number, the length ratio being constant. Changing the values of D_B and D_P, but keeping their ratio constant, is equivalent to changing all dimensions in the same proportion; *i.e.*, changing the *size* but maintaining the same *shape*. This is exactly the condition of geometric similarity; therefore, if all of the plenum, take-off, and branch duct combinations to be studied were geometrically similar, λ would depend only on the velocity ratio and Reynolds number. Referring to Fig. 2, it can be shown that the condition of geometric similarity will not be fulfilled during this investigation. Note that although the diameters of the take-offs vary from 7 in. to 9 in., certain dimensions are the same for all take-offs. Their shapes are therefore different, and the take-offs are not geometrically similar. Moreover, in

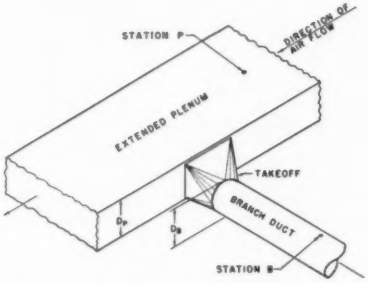


FIG. 3. VIEW OF AN EXTENDED PLENUM, SIDE TAKE-OFF, AND BRANCH DUCT

describing the apparatus it is stated that the plenums used in this investigation were of the same width but of different depths. Therefore, the plenums are also not geometrically similar. It can therefore be concluded that, because of the variable length ratios that are to be used in the investigation, the effect of length ratio on λ should be determined experimentally.

This analysis shows that the six variables influencing λ, as given in Equation 7, can be reduced to the set of three dimensionless variables given in Equation 9. The scope of the experimentation can therefore be limited to a study of each one of the three variables in Equation 9 while holding the other two constant. By this method, the extent of the actual significance of each variable can be determined.

RESULTS OF STUDIES WITH A SINGLE TAKE-OFF

The initial studies were conducted with single take-offs installed on the 24-in. × 8-in. plenum. When compiling the laboratory data, it was found that the independent loss coefficient, λ, varied considerably with the velocity ratio. Consequently, this ratio was usually used as the basis for expressing the experimental results.

Side Take-offs: The relationship between independent loss coefficient and velocity ratio for Take-off A is shown in Fig. 4 for three plenum sizes. Curves 1 and 2 were obtained from studies conducted with the 24-in. × 8-in. and 24-in. × 12-in. plenums. Curve 3 was replotted from a previous paper[1] which reported the performance of Take-off A when installed on a 12-in. × 8-in. plenum.

A comparison of curves 1 and 3 shows the effect of *plenum width* on the independent loss coefficient. Although it appears from the curves that the plenum

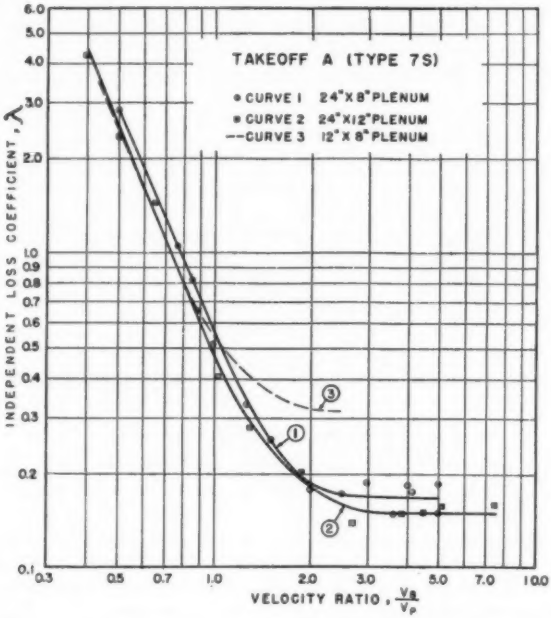


FIG. 4. PERFORMANCE OF A SIDE TAKE-OFF ON THREE SIZES OF PLENUMS

width has a marked effect on the independent loss coefficient at high velocity ratios, the maximum difference between curves 1 and 3, which occurs at a velocity ratio of 2.5, corresponds to a difference in pressure loss of less than four equivalent feet of 7-in. diameter duct. For purposes of system design, this difference is negligible; consequently, it can be concluded that increasing the width of the plenum from 12 in. to 24 in. has no significant effect on the independent loss coefficient of Take-off A.

Comparison of curves 1 and 2 shows the effect of *plenum depth* on the independent loss coefficient. The maximum difference between these curves, which occurs at velocity ratios less than one, corresponds to a pressure loss of less than five equivalent feet of 7-in. diameter duct. For the purpose of design, this

difference is negligible; therefore, it can be concluded that the increase in depth from 8 in. to 12 in. had no effect upon the independent loss coefficient of Take-off A. Take-offs B and C, which were similar in size and shape to Take-off A except for slight variations in manufacture, were also studied. The results showed that the individual take-off losses expressed in terms of equivalent length were nearly the same as those for Take-off A. Moreover, studies on the effect of

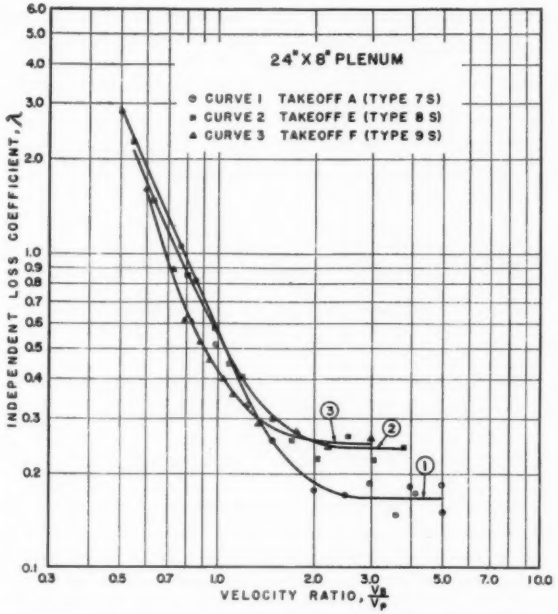


FIG. 5. PERFORMANCE OF THREE SIZES OF SIDE TAKE-OFFS

plenum depth conducted with Take-offs B and C confirmed the conclusion stated for Take-off A.

Similar studies on the effect of plenum depth were made for Take-offs E and F. The results showed that an increase in depth from 8 in. to 12 in. had no effect on the independent loss coefficients of Take-offs E and F.

The relation between the independent loss coefficient and the velocity ratio for three sizes of side take-offs with the 24-in. × 8-in. plenum is shown in Fig. 5. These curves, which are for take-offs of different sizes and of slightly different shapes, are in good agreement with each other. Although the maximum difference between any two curves corresponds to 10 equivalent feet of round duct, this is small in comparison to the equivalent length of the entire duct system.

In the preliminary analysis it was stated that the effect of either velocity ratio, length ratio, or Reynolds number could be determined by holding any two of these variables constant and varying the third. In order to investigate the influence of Reynolds number on λ, an analysis was first made of the laboratory data obtained with Take-off A installed on the 24-in. × 8-in. plenum, since one configuration of the apparatus corresponds to a constant value of length ratio D_B/D_P. For selected values of velocity ratio V_B/V_P, values of the independent loss coefficient were plotted against the Reynolds number in the branch duct. Representative results are shown by the circled points and curves in Fig. 6. Since the curves are horizontal lines for a given velocity ratio, the independent loss coefficient is a constant. The significant conclusion can therefore be drawn that, within the limits of experimental accuracy, λ is independent of Reynolds

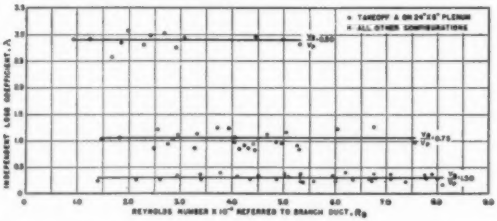


FIG. 6. PERFORMANCE OF TAKE-OFFS AS RELATED TO REYNOLDS NUMBER

number for the range over which data were obtained with this configuration of the apparatus.

An analysis was next made of the data obtained for all other configurations of the apparatus. For each configuration and a constant value of velocity ratio, values of the independent loss coefficient fell above and below the corresponding curves in Fig. 6. Since no trend could be established, all of the points have been plotted in black filled circles in the figure. The agreement of these points with the previously established curves permits two important conclusions to be made. *First,* for each configuration of the apparatus, λ is independent of Reynolds number. *Second,* for a given value of velocity ratio, a single curve applies within the limits of experimental accuracy to all configurations of the apparatus. Since the various side take-offs and plenums were not geometrically similar, the latter conclusion is equivalent to a statement that the effect of the deviations from geometrical similarity are insignificant. The length ratio and Reynolds number can therefore be discarded from Equation 9. The result is then

$$\lambda = \psi_S(V_B/V_P) \quad \ldots \ldots \ldots \ldots \ldots \quad (10)$$

in which the symbol ψ_s indicates a general functional relationship applicable to side take-offs. Equation 10 expresses a unique relationship between λ and V_B/V_P that holds with satisfactory accuracy for the configurations studied and for the practical range of air flow rates. Therefore, a single curve relating λ to V_B/V_P can be used to express all of the experimental results obtained with side take-offs.

It is of particular significance that the curves of Fig. 6 are in good agreement with those obtained by other investigators for a similar 7-in. diameter side take-off and a 12-in. × 8-in. extended plenum[1].

Top Take-offs: The results of studies conducted with a single 7-in. diameter top take-off installed on the 24-in. × 8-in. plenum are shown in Fig. 7, together with the results for an 8-in. diameter top take-off under the same experimental

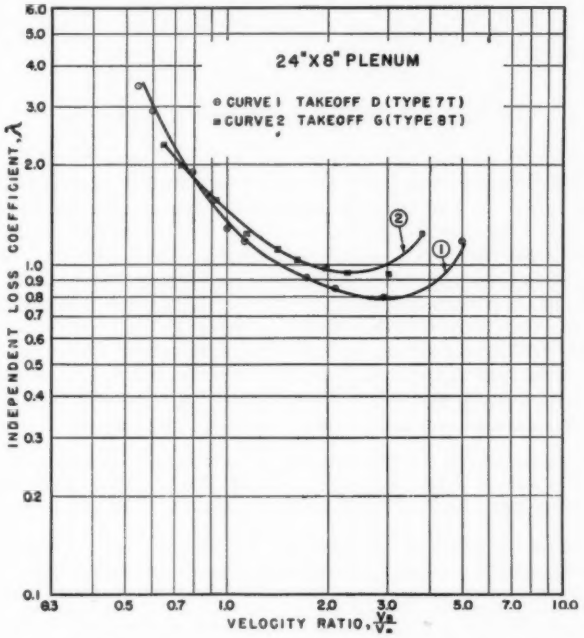


FIG. 7. PERFORMANCE OF TWO SIZES OF TOP TAKE-OFFS

conditions. The 8-in. diameter take-off, Take-off G, has the higher independent loss coefficient over most of the velocity ratio range. However, for practical considerations, the agreement between the two curves is good since the maximum difference between them represents less than 10 equivalent feet of round duct. Comparison of Figs. 5 and 7 shows that top take-offs have much higher pressure losses than side take-offs for the same experimental conditions. This is to be expected because the air flowing through the top take-off must make two 90° turns in passing from the plenum into the branch duct; whereas, the air flowing through the side take-off must make but a single 90° turn.

Take-offs D and G were also studied independently on the 24-in. × 12-in. plenum. As the results were in excellent agreement with the curves of Fig. 7,

it was concluded that an increase in plenum depth from 8 in. to 12 in. had no significant effect on the independent loss coefficients of 7-in. and 8-in. diameter top take-offs.

An analysis of the effect of Reynolds number and length ratio was also made with the data obtained for the top take-offs. As with the side take-offs, it was found that the laboratory results could be expressed by a unique relationship of the form

$$\lambda = \psi_T(V_B/V_P) \quad \ldots \quad (11)$$

in which the symbol ψ_T indicates a general functional relationship applicable to top take-offs.

RESULTS OF STUDIES WITH MULTIPLE TAKE-OFFS

Studies were conducted with two or more take-offs installed on the 24-in. × 8-in. extended plenum in order to determine the effect of the relative location of the take-offs. Each of the loss coefficients determined during these studies was designated the *interference loss coefficient* θ. The difference between the independent loss coefficient λ and the interference loss coefficient θ was designated the *take-off interference*. The take-off interference serves as an index of the change in the performance of a take-off due to its location in relation to the locations of other take-offs. During these studies, values of the take-off interference were obtained over the practical range of velocity ratios by adjusting the dampers on the various branch ducts and at the exit section of the plenum.

The number, types, and relative locations of the take-offs for each of the various arrangements studied are shown in Fig. 8, as follows:

1. *Arrangement in Fig. 8a:* These studies were conducted with two 7-in. diameter side take-offs located along the same side of the plenum. Take-off B was initially installed 67-in. downstream from Take-off A, and was then moved progressively closer to Take-off A. The maximum take-off interference for either take-off, expressed in terms of equivalent feet, was two feet of round duct. Therefore, it was concluded that there was negligible interference between two 7-in. diameter side take-offs due to their relative locations on the same side of the extended plenum.

2. *Arrangement in Fig. 8b:* Take-off A was fixed in place, and Take-off B was located at a distance of 32 in. or 16 in. downstream and on the opposite side of the plenum. The results showed that the performance of the downstream take-off was affected by the proximity of the up-stream take-off. However, the take-off interference was only of the order of five equivalent feet of round duct. In contrast, the performance of the upstream take-off was not influenced by the relative locations of the downstream take-off.

3. *Arrangement in Fig. 8c:* When Take-offs A and B were located directly opposite each other, the maximum take-off interference for either take-off was only four equivalent feet of round duct.

4. *Arrangement in Fig. 8d:* In order to determine the effect of take-off size on the take-off interference, Take-offs E and F, which were 8 in. and 9 in. in diameter, respectively, were studied when located directly opposite each other. The take-off interference was greater than that for the 7-in. diameter take-offs shown in Fig. 8c. However, the take-off interference for Take-off E or Take-off F did not exceed nine equivalent feet of round duct.

5. *Arrangement in Fig. 8e:* This is similar to the arrangement shown in Fig. 8c except that an additional 7-in. diameter take-off was installed 32-in. downstream from Take-off B. The performance of the downstream take-off was not adversely affected by the presence of two take-offs upstream since the maximum take-off interference was equivalent to three feet of round duct.

6. *Arrangement in Fig. 8f:* From the standpoint of space limitations, the largest number of take-offs which would ordinarily be located on the plenum at a given cross section would be two side take-offs and one top take-off. This arrangement was studied with three 7-in. diameter take-offs. The results showed that, as before, the

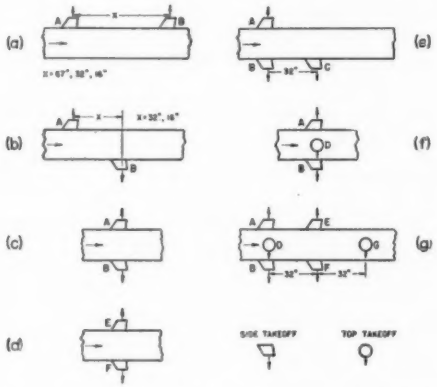


FIG. 8. ARRANGEMENTS FOR STUDIES OF MULTIPLE TAKE-OFFS

side take-offs did not interfere with each other. However, over the practical range of velocity ratios, the interference loss coefficient θ of the top take-off was *less* than its independent loss coefficient λ as shown in Fig. 9. The maximum effect of take-off interference as indicated in Fig. 9 represents a reduction of 13 equivalent feet of round duct. For practical purposes, the independent loss coefficient λ should be used and the safety factor represented by the *negative* take-off interference ignored.

7. *Arrangements in Figs. 8c, 8d, and 8f:* Studies were also conducted with the 24-in. x 12-in. plenum in order to determine the effect of plenum depth on take-off interference. No appreciable change in take-off interference resulted from the increase in plenum depth.

8. *Arrangement in Fig. 8g:* Studies were made with the 24-in. × 8-in. plenum in order to determine the interference of each take-off in a simulated practical system. The interference of the various take-offs was found to be in excellent agreement with those previously determined. In addition, the interference effect was not cumulative when a take-off was located downstream from several others. Studies were also made with the 24-in. × 12-in. plenum, and similar results were obtained.

These studies with multiple take-offs indicate that the independent loss coefficient defines the pressure loss of a take-off, and can be used in predicting the

air flow rate of the branch duct. To verify this statement, special studies were conducted with the 24-in. × 8-in. plenum and with the simulated practical system in Fig. 8g. A value of equivalent length was arbitrarily selected for each branch duct. The air flow rate in each branch duct was then predicted using the independent loss coefficients of the various take-offs. Next, the actual air flow rates in the branch ducts were measured and compared with the predicted

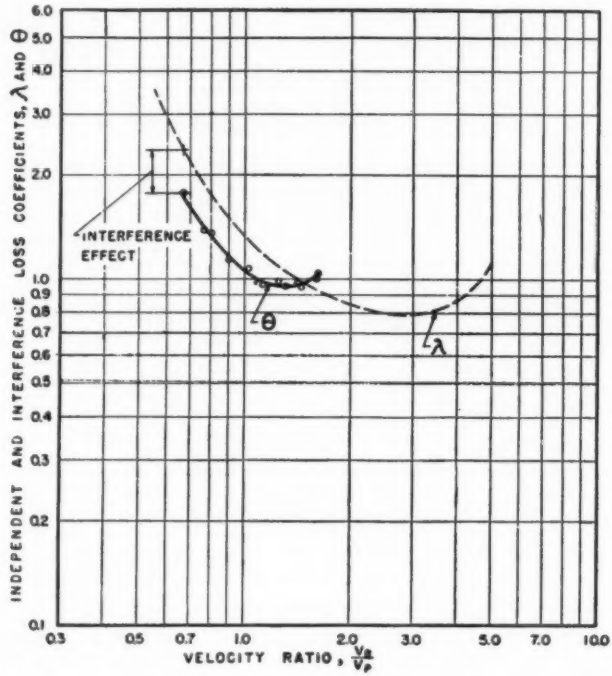


FIG. 9. THE EFFECT OF INTERFERENCE ON A TAKE-OFF

values. In no case did the actual branch duct flow deviate from the predicted value by more than five percent and the average deviation was only two percent. Deviations of the same order were obtained during a similar study with the 24-in. × 12-in. plenum.

The results obtained from these studies with multiple take-offs show that interference due to the relative location of take-offs is negligible from a practical standpoint. Hence, there are no limitations on the relative location of the take-offs. Furthermore, the results show that the air flow rate in a branch duct can be predicted by means of the independent loss coefficient as determined by the velocity ratio at the point of take-off.

PRACTICAL INTERPRETATION OF RESULTS

In the design of air duct systems it is usual practice to consider the pressure loss of a take-off in terms of *equivalent feet of round duct* or *equivalent diameters* rather than in terms of the dimensionless independent loss coefficient, A. Furthermore, it is customary to use a single value of equivalent length to represent the loss of a take-off for systems such as the trunk duct system, whereas the results of this investigation have shown that this is not feasible for extended-plenum take-offs because the loss is dependent upon the velocity ratio. It is therefore desirable to express the performance of extended-plenum take-offs in terms of the equivalent feet of round duct as a function of velocity ratio.

In the preceding discussion, it was indicated that for *practical considerations* the performance of extended-plenum take-offs when expressed in terms of equivalent feet were not affected by the following:

1. Changes in the width of the extended plenum.
2. Changes in the depth of the extended plenum.
3. Variation in the size of side take-offs.
4. Variation in the size of top take-offs.
5. Proximity of other take-offs.

These factors indicate the possibility of establishing a single curve applicable to the side take-offs and another curve applicable to the top take-offs. Further analysis of this possibility proved that such curves could be drawn, and they are presented in Fig. 10. In no case does the actual pressure loss of any side or top take-off, as determined experimentally, exceed by more than five equivalent feet the loss obtained from the corresponding curve of Fig. 10. Since five equivalent feet of branch duct represent but a small fraction of the normal pressure loss of the complete duct system, it can be concluded that these curves can be utilized in designing extended-plenum duct systems having plenum and take-off sizes of the order of those investigated. The experimental results presented in Figs. 4, 5, 7, and 9 have been condensed in Fig. 10 to two curves relating take-off loss to velocity ratio.

Certain practical aspects can be illustrated using the curves of Fig. 10. Referring to the curve for side take-offs, the loss varies from 80 to 7 equivalent feet of branch duct, the larger loss being at the lower range of velocity ratios. For example, a branch velocity of 400 fpm and a plenum velocity of 800 fpm represent a velocity ratio of 0.5 and results in a take-off loss of 80 equivalent feet. In contrast, a branch velocity of 400 fpm and a plenum velocity also of 400 fpm represent a velocity ratio of 1.0 and a take-off loss of only 15 equivalent feet.

Referring to the curve for top take-offs in Fig. 10, the loss varies from 70 to 25 equivalent feet over the range of velocity ratios investigated, the larger loss being at the lower range of velocity ratios. It should be noted that at a given value of velocity ratio, the pressure loss of a top take-off is considerably greater than that of a side take-off. Therefore, top take-offs should be located at points on the plenum where a relatively high velocity ratio exists.

To determine when low velocity ratios, and therefore high take-off losses, occur, consider the characteristics of an extended plenum system. A certain

volume and velocity exist in the plenum just before the first take-off. A portion of the air in the plenum then flows into this take-off and thence into the branch duct, and the remainder continues down the plenum. As a result, the volume flowing in the plenum just after the take-off is less than that in the plenum just before the take-off. Since the area of the plenum is constant throughout its length, the plenum velocity decreases after each take-off. Current residential practice is to use design velocities of from 700 to 900 fpm for the sections of the plenum close to the conditioner.[3] The plenum velocity may therefore be as high as 900

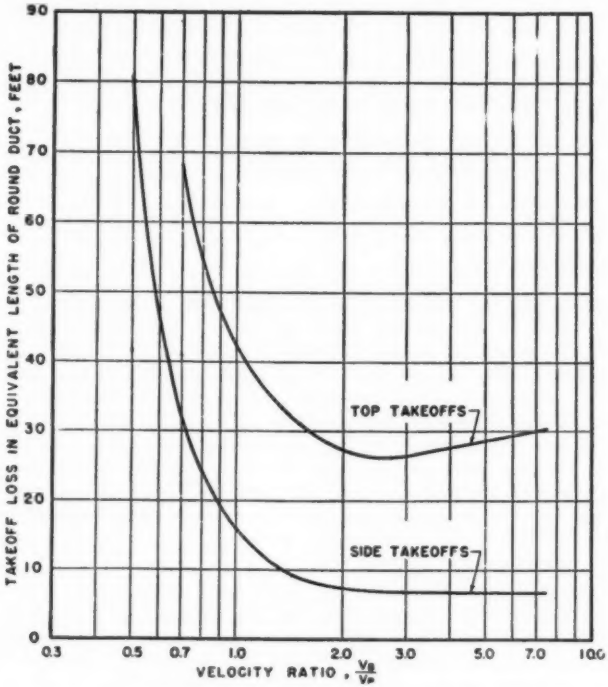


FIG. 10. LOSS CURVES FOR SIDE AND TOP TAKE-OFFS

fpm at the first take-off and 100 fpm or less at the last take-off. Since the design velocity for branch ducts of residential systems are generally of the order of 600 fpm,[3] it is evident that low velocity ratios and large losses are characteristic of the first few take-offs, and high velocity ratios and small losses are characteristic of the last few take-offs on an extended plenum. This indicates that during the design process careful consideration should be given to the first few take-offs.

Whenever possible, velocity ratios for side take-offs should be greater than 0.70 in order that the resulting take-off loss will not exceed 35 ft of round duct.

In addition, velocity ratios for top take-offs should be greater than 1.0 to provide losses less than 45 ft of round duct. If, during the design of an extended plenum system, these conditions of velocity ratios are not met, one or more of the following changes in the system should be considered:

1. Increase the plenum size, to decrease the plenum velocity. This provides a higher velocity ratio and a smaller take-off loss.

2. Increase the branch duct velocity by increasing the air flow rate in the branch, or by decreasing the size of the branch duct. This may require a slightly higher pressure in the plenum.

3. Rearrange the layout so that the branch ducts having low equivalent lengths and large volume requirements will be connected to the first few take-offs. In this regard, the location of a top take-off on the plenum at a point close to the conditioner should be avoided, since the velocity ratio will tend to be low at this point.

From information already available, it is possible to determine the pressure losses of the various components of extended-plenum duct systems[3] except those of the take-offs. In this paper the pressure losses of standard side and top take-offs can be evaluated from the curves of Fig. 10, and, therefore, sufficient information is now available for designing extended-plenum duct systems for residential and small commercial applications.

CONCLUSIONS AND RECOMMENDATIONS

For the range of take-offs and extended plenums studied in this investigation, the following conclusions and recommendations can be made:

1. The independent loss coefficient, λ, is a function of velocity ratio, but is not a function of length ratio and Reynolds number.

2. The independent loss coefficient of a side take-off is not affected by the size or shape of the extended plenum, the size of the take-off, or the location on the plenum relative to other take-offs. This statement also applies to a top take-off.

3. For a given velocity ratio, the pressure loss of a top take-off is much greater than that of a side take-off.

4. The pressure losses of both side and top take-offs vary over a wide range, and are exceedingly large for low values of velocity ratios. Therefore, low values of velocity ratio should be avoided whenever possible.

5. For practical purposes, the relationship between pressure loss and velocity ratio is given by a single curve that applies to all side take-offs. Another curve provides this relationship for all top take-offs.

6. Careful consideration should be given to the design of branch ducts located near the conditioner in view of the possibility of obtaining low values of velocity ratio and resulting high values of take-off loss.

7. In order to avoid excessive take-off losses, the branch duct size, take-off location, and plenum size should be selected to maintain values of velocity ratio greater than 0.7 for side take-offs and greater than 1.0 for top take-offs.

ACKNOWLEDGMENT

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systems for all-year gas air conditioning. The results of research previously completed under the terms of this agreement have been reported.[4, 5, 6] Acknowledgment is made to the A.G.A. Technical Advisory Group for Heating and Air Conditioning Research for the advice and counsel received.

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1431

No. 1431

WARM-AIR PERIMETER HEATING—HEAT EMITTED FROM FLOOR SURFACE†

PART I

By J. R. JAMIESON*, R. W. ROOSE** AND S. KONZO††, URBANA, ILL.

ONE MEANS for providing for low-cost homes has been the construction of basementless structures in which the concrete floor slab is laid on the ground. Such floors are likely to be cold[1, 2] unless some provision is made for adequate insulation[3] of the edge of the floor slab and for introduction of some heat into the floor, especially near the edges where the greatest heat loss occurs.

The warm-air perimeter system[4] was devised to eliminate the cold floor at the exposed edge by the application of heat in the peripheral area of the floor. In this system the warm air from the furnace is delivered through sub-floor feeder ducts extending from a plenum chamber under the furnace to the perimeter duct, which is embedded in the floor along the outer walls. A panel heating effect (by radiation and convection) is provided by the warm floor above the perimeter duct, and convection heat is supplied by the warm air which is introduced into the rooms through floor registers located along the outer walls. Hence, all of the heated air leaving the furnace is delivered into the rooms through registers and is returned to the furnace through return-air intakes located at the inside walls or ceiling.

The overall performance of one type of perimeter heating system has been studied[5] in the Warm Air Heating Research Residence No. 3 at the University of Illinois. Since the many variables affecting the heat transfer from the embedded ducts to the room and to the ground could not be studied individually in the Research Residence it was considered desirable to conduct a separate

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† This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*, and is the first of a series dealing with a reesarch investigation conducted in the Floor Slab Laboratory at the University of Illinois.

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[1] Exponent numerals refer to References.

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investigation of these various factors over a two-year period in the Floor Slab Laboratory. The overall objectives of the investigation were to determine:

1. The magnitudes of, and decreases in temperature of the air flowing through both the embedded perimeter and feeder ducts.

2. The magnitudes and distribution of the heat losses to the ground and through the edges of the floor slab.

3. The distribution and the magnitude of floor surface temperatures, as well as the heat emission from the floor over the duct (panel effect).

4. The air distribution and the room-air temperature gradients resulting from the use of various types of floor and baseboard registers located along the exterior wall.

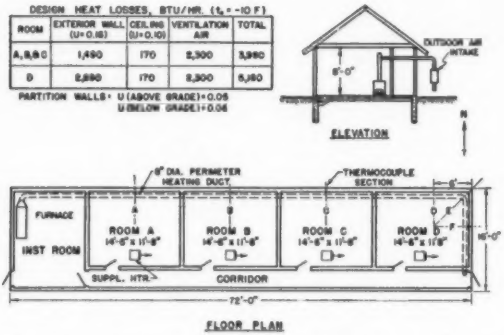

FIG. 1. ARRANGEMENT OF EXPERIMENTAL ROOMS IN FLOOR SLAB LABORATORY

5. The temperatures and moisture contents of the underlying soils as well as the effect of soil moisture on edge insulation.

6. The effect of a floor covering on Items 1, 2, and 3.

7. The change in floor surface, room-air, and mean radiant temperatures resulting from the use of different sized windows.

For each of the foregoing objectives a number of independent variables existed or were separately introduced. These variables included the following items:

1. Construction of the floor slabs with four different types of edge insulation.

2. The temperatures and velocities of the circulating air in the perimeter duct.

3. Weather effects, such as outdoor temperature, solar intensity, wind velocity and direction, and precipitation.

In this paper the discussion includes only Item 3 and that part of Item 6 of the overall objectives referring to heat transfer at the floor surface.

DESCRIPTION OF FLOOR SLAB LABORATORY

The Floor Slab Laboratory, shown in Fig. 1, was constructed in 1947 for the specific purpose of studying methods of insulating unheated concrete floor slabs.[3] The one-story building was modified in 1949, to accommodate concrete floor slabs with embedded perimeter heating ducts. Further modification was made the

following year to provide for the study of objectives 4 and 7. Rooms A, B, and C were exposed only to the north while Room D was exposed to both north and east; thus, the effect of direct solar radiation upon the rooms was minimized. A summary of the overall heat transfer coefficients and design heat losses for each room is also given in Fig. 1. The insulated partition walls between rooms extended 4 ft below the floor to minimize any heat transfer from adjoining spaces.

The four types of floor-slab edge construction investigated during the 1949-50 and the 1950-51 heating seasons are shown in Fig. 2. The edge insulation used consisted of a rigid and waterproof cellular glass, having a thermal conductivity of 0.42 Btu per (hr) (sq ft) (F deg per in.). A 4 in. layer of coarse gravel under the floor provided suitable drainage and minimized the capillary rise of

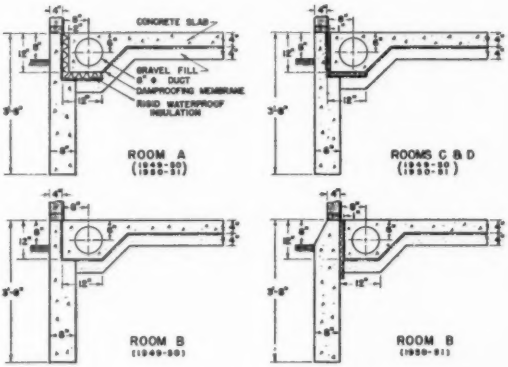

FIG. 2. TYPES OF EDGE CONSTRUCTION

moisture from the soil. A duplex-kraft paper was placed over the gravel fill for a membrane damproofing, all the joints being lapped and then sealed with cold tar. The soil under and adjacent to the Laboratory contained 3 ft of fill which was primarily a silty clay composed of 18 percent clay, 53 percent silt, 24 percent sand and 5 percent gravel. Under the fill was the original black silty-clay topsoil which extended to a depth of 4 ft. The soil adjacent to the test rooms was graded 8 in. below the floor line and sloped away from the building for proper drainage.

An electric furnace located in the instrument room supplied warm air to the perimeter duct. For the studies reported in this paper, warm air was not introduced into the rooms from the perimeter duct as is the normal procedure in a residence provided with a perimeter heating system.[4] A supplemental electric heater, controlled by a thermostat, was placed in each room to supplement the heat gains from the perimeter duct and to simulate the heat input through a conventional register. With this arrangement, a constant air temperature of 72 F was maintained in each room at the 30-in. level. A mixture of 20 cfm of outdoor air, corresponding to one air change per hour, and 60 cfm of recirculated air were continuously passed through the supplemental heater. A small flexible

damper in each door allowed air to leave the room. The corridor and instrument room were also maintained at 72 F by separate electric heaters. Similar heaters were used in the attic to maintain a temperature of 60 F.

Complete instrumentation for temperature measurement was provided, consisting of approximately 600 thermocouples made of No. 24 B and S gage copper and constantan wires. In addition to the thermocouples used to measure the temperatures of room-air, duct-air, floor surface, etc., a large number of thermocouples were placed in the soil and in the gravel and concrete under each

TABLE 1—STUDIES MADE WITH HEAT-FLOW METERS AND THERMOCOUPLES IN ROOM A

TIME OF TEST		TEST SERIES	AVG DUCT-AIR TEMP F	AVG DUCT-AIR VELOCITY (FPM)	AVG OUTDOOR TEMP F	NO. OF HEAT-FLOW METERS INSTALLED
Starting Date	Total No. of Hr					
Jan. 11, 1950	72	ABCD-3	134	590	34.5	3
Jan. 31, 1950	72	ABCD-4	157	610	24.8	3
Mar. 2, 1950	24	ABCD-6	106	915	15.0	8
Mar. 24, 1950	24	ABCD-7	106	375	41.6	8
Mar. 29, 1950	24	ABCD-8	106	375	31.0	8
11:00 A.M. Apr. 11, 1950	0[a]	ABCD-9	107[b]	375	39.0	8
12:30 P.M. Apr. 11, 1950	0[a]	ABCD-9	164[b]	375	40.5	8
5:00 P.M. Apr. 12, 1950	0[a]	ABCD-9	185[b]	375	33.0	8
5:00 A.M. Apr. 14, 1950	0[a]	ABCD-9	188[b]	375	25.5	8
2:00 A.M. Apr. 15, 1950	0[a]	ABCD-9	75	375	29.0	8
Jan. 23, 1951	192	A-13	145	150	15.1	8
Feb. 1, 1951	144	A-13a	145	150	15.5	8

[a] Data taken with continuous recorder during transient heat-flow test. During this test the duct-air temperature was suddenly increased, allowed to level off and then rapidly decreased.
[b] Instantaneous temperatures.

room in a section perpendicular to the axis of the perimeter duct as designated in Fig. 1.

Fifteen heat-flow meters of the Boelter type[6,7] were used for measuring the flow of heat to or from the floor surface, eight of these being located in Room A for a detailed study of heat transfer at the floor surface.

Instrumentation was also provided to measure the relative humidity, the electrical input to the supplemental room heaters, the duct-air velocities, the soil moisture contents, and the level of the water table under the building.

PROCEDURE

The primary purpose of the study reported in this paper was to determine the floor surface temperature distribution and the rate of heat emission to the room by radiation and convection from various parts of the floor surface. The

initial phase of this study was made on bare concrete floors. A secondary study of the effect of a floor covering on the heat transfer from the floor surface was conducted in Room A.

Four experimental and analytical methods were available for determining the rate of heat flow from the floors as follows:

Method A. By empirical heat transfer equations—utilizing the measurements of room-air, floor, ceiling and wall surface temperatures and applying them to equations for radiation and natural convection.

Method B. By direct measurement—using heat-flow meters.

Method C. By applying a heat balance—equating the heat gain from the floor surface and supplemental heaters with the above-ground heat losses.

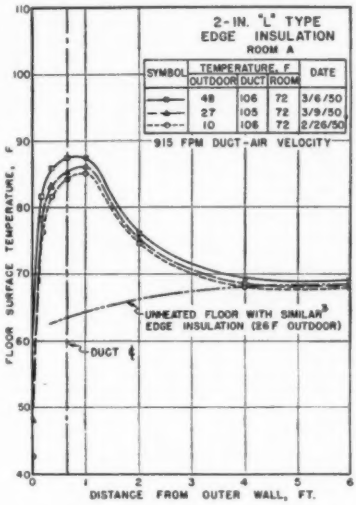

FIG. 3. FLOOR SURFACE TEMPERATURE PATTERNS WITH UNHEATED AND PERIMETER HEATED FLOORS

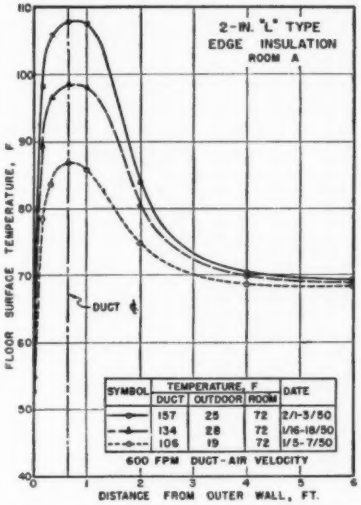

FIG. 4. EFFECT OF DUCT-AIR TEMPERATURES UPON FLOOR SURFACE TEMPERATURE PATTERN

Method D. By heat-flow diagrams—graphically analyzing the temperature pattern in the slab and ground. The method was found to give a qualitative analysis, but the data were not sufficient to give precise quantitative values of heat flow.

Continuous records of outdoor, room-air, duct-air, and other critical temperatures were made throughout the investigation. Daily readings were taken of all thermocouples, heat-flow meters, and duct-air velocities; in addition, the special studies listed in Table 1 were made when relatively stable weather prevailed.

FLOOR SURFACE TEMPERATURE DISTRIBUTION

Basementless homes built with concrete floor slabs laid directly on the ground are subject to cold floors, particularly around the periphery of the house where

the heat loss from the floor slab is greater than that through the center portion of the floor. Typical patterns of the floor surface temperatures, both with the floor heated and unheated, are shown in Fig. 3. The highest floor surface temperature was observed over the center of the perimeter duct. Furthermore, the floor surface temperatures in the peripheral area within four feet from the outer wall were markedly increased by the presence of the heated perimeter duct. The temperature patterns of the heated floor such as shown in Fig. 3 are affected by a number of factors of which the following four are of major importance:

1. *Outdoor Temperature:* For a constant duct-air velocity and temperature, the change in floor surface temperatures resulting from a change in outdoor temperature

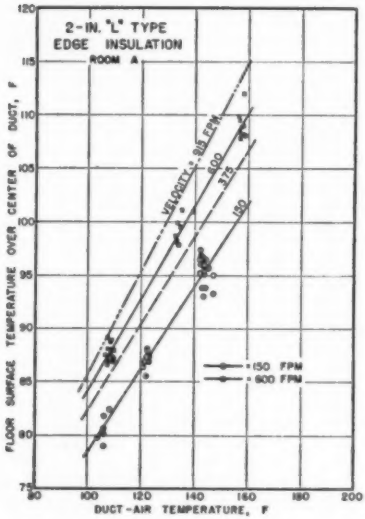

FIG. 5. EFFECT OF DUCT-AIR VELOCITY UPON FLOOR SURFACE TEMPERATURE OVER CENTER OF DUCT

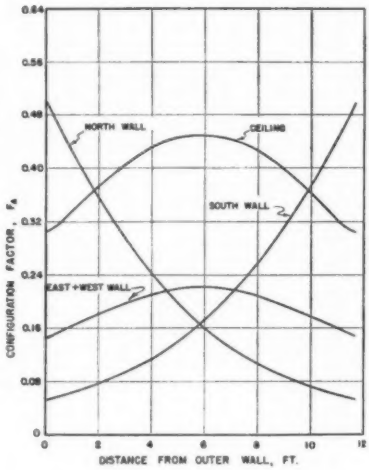

FIG. 6. CONFIGURATION FACTOR, F_A, FOR A LINE ON THE FLOOR SURFACE PERPENDICULAR TO THE CENTER OF THE OUTER WALL

was small. For example, as shown in Fig. 3, the difference in floor surface temperatures over the center of the duct was only 3 deg for a difference in outdoor temperature of 38 deg. Smaller differences in floor surface temperatures were observed in the center section of the floor. The lower surface temperature of the floor accompanying the lower outdoor temperature has been attributed to an increase in radiation heat transfer from the floor to the cooler surrounding surfaces. However, it should be noted that in an actual residence this reduction in floor surface temperature would not occur since the increased heat demand in cooler weather would automatically bring about an increase in duct-air temperature.

2. *Duct-Air Temperature:* For a constant duct-air velocity and outdoor temperature the floor surface temperatures were markedly affected by the duct-air temperature. For example, the surface temperatures over the center of the duct, as shown in Fig. 4, increased 20 deg when the duct-air temperature was increased by 50 deg. The maximum floor surface temperature of 108 F was obtained with a duct-air tem-

perature of 157 F. In this connection the recommended values for maximum allowable surface temperatures[8] are 120 F for borders and aisles and 85 F for the remainder of the floor.

3. *Edge Insulation:* Previous investigations[1, 3] made with unheated floors indicated that the use of edge insulation increased the temperature of the floor surface immediately adjacent to the outer wall, but was relatively ineffective as far as the floor surface in the middle of the room was concerned. A similar trend was observed in the present investigation with the floor heated by a perimeter duct.

4. *Duct-Air Velocities:* A large number of observations of floor surface temperatures over the perimeter duct were made with various values of the duct-air temperature. The average curves shown in Fig. 5 were based upon these observed data. In order to avoid confusion, only the observed values for 150 fpm and 600 fpm velocity have been included. A large portion of the scatter of points can be accounted for by the fact that constant outdoor temperatures did not prevail. In general, the curves can be considered to represent an average outdoor temperature of 25 F. The fact that higher floor surface temperatures were obtained with the larger values of duct-air velocities is in agreement with the predicted trends based upon considerations of heat transfer, that is, an increase in duct-air velocity would decrease the thermal resistance of the air film adjacent to the duct surface, and hence would result in a higher floor surface temperature as well as a higher rate of heat transfer to the room.

DETERMINING RATE OF HEAT FLOW FROM FLOOR

A considerable disagreement exists in the literature concerning the magnitude of the total heat emission from a heated floor slab. Although the floor slab heated with a perimeter duct exhibits temperature characteristics which differ from those of a fully heated floor, it should be possible to evaluate the heat emission characteristics and thereby to ascertain the validity of the claims made in regards to the magnitude. For this purpose, one analytical method, designated as Method A, and two experimental methods, designated as Methods B and C, were used. The results obtained by each method will be discussed.

METHOD A—THE APPLICATION OF TEMPERATURE MEASUREMENTS TO EMPIRICAL HEAT TRANSFER EQUATIONS

Method A involves the use of semi-empirical equations of heat transfer. Part 1 of this discussion refers to the radiation component, Part 2 to the convection component of heat transfer, and Part 3 to the total heat emission from the floor. In this study, the heat flow from the floor surface was considered per unit area of floor surface along the center line of the room, perpendicular to the exposed wall.

1. Radiation: The transfer of heat by radiation can be expressed by the Stefan-Boltzmann equation:

$$(q_r/A) = 0.173 \times 10^{-8} F_E F_A (T_s^4 - T_a^4) \quad \ldots\ldots\ldots\ldots (1)$$

where

(q_r/A) = heat loss by radiation to or from the floor, Btu per (hour) (square foot).
F_E = the emissivity factor, dimensionless.
For a large enclosed body, or parallel planes,
$F_E = 1/(1/e_1 + 1/e_2 - 1)$
where e_1 and e_2 are the emissivities of the emitting and absorbing surfaces, respectively[9]. All surfaces were assumed to have an emissivity of 0.90, or a combined effective emissivity, F_E of 0.82.

F_A = the configuration factor, dimensionless. This factor accounts for the shapes and relative positions of the emitting and absorbing surfaces.
T_e = temperature of the emitting surface Fahrenheit, absolute.
T_a = temperature of the absorbing surface Fahrenheit, absolute.

The heat transfer by radiation can be calculated by considering the radiation exchange from the floor to each unheated surface taking into account the shapes and relative positions of the surfaces involved in the exchange. In this procedure, each unit area of the floor is considered to be exposed to a mean radiant temperature which in turn is dependent upon the temperature of each of the surrounding surfaces and the included solid angle which the surface makes with respect to the unit area. This procedure is hereafter designated as the mean radiant temperature, or MRT, method. The heat emission by radiation from a unit area of the floor to the walls and ceiling can be expressed by the summation:

$$(q_r/A) = (q_{rn}/A) + (q_{re}/A) + (q_{rs}/A) + (q_{rw}/A) + (q_{rc}/A) \quad (2)$$

The subscripts n, e, s, w, c, refer to the respective north wall, east wall, south wall, west wall, or ceiling surface. In terms of Equation 1 this becomes:

$$(q_r/A) = 0.173 \times 10^{-8} \times 0.82 \left[F_{An}(T_f^4 - T_n^4) + F_{Ae}(T_f^4 - T_e^4) + F_{As}(T_f^4 - T_s^4) + F_{Aw}(T_f^4 - T_w^4) + F_{Ac}(T_f^4 - T_c^4)\right] \quad (3)$$

The temperature, T_f, was the floor surface temperature of each unit area. The configuration factor F_A must be evaluated for each unit area of the floor with respect to each separate wall or ceiling surface. The values shown in Fig. 6 were determined by the graphical method of Hottel,[9] and were used with Equation 3 for the determination of the radiant heat transfer component of the total heat flow at the floor surface.

The preceding MRT method involves tedious computations so that a simpler procedure, designated herein as the AST method, is generally used. In this procedure the temperature of the absorbing surface T_a is assumed to be equal to the weighted average surface temperature (AST) of the walls and ceiling. That is,

$$(\text{AST}) = (t_n A_n + t_e A_e + t_s A_s + t_w A_w + t_c A_c)/(A_n + A_e + A_s + A_w + A_c) \quad (4)$$

where

t = temperature of the unheated surface, Fahrenheit.

The value of AST thus obtained and converted to absolute temperatures replaces the temperature T_a in Equation 1. Also, a value of 1.0 is used for the configuration factor F_A.

Both the MRT and the AST methods were applied to a specific condition for which the surface temperatures are shown in Fig. *7a*. The resulting radiation components of heat transfer from the floor are shown in Fig. *7b*. In general, the MRT method produced higher results than those by the AST method for the section of the floor studied. The largest deviation, which was only 2 Btu per (hr) (sq ft), occurred directly over the center of the perimeter duct. The deviation can be attributed to the fact that in the MRT method the large solid angle generated by the cold outer wall was taken into account, whereas in the AST method the temperature of the cold north wall was given the same weight as that for the warmer south wall and the configuration factor of both walls

was considered to be the same. Since the MRT method gives precise results, it was used throughout this analysis.

2. Convection: The convection component of total heat emission from the floor can be determined from semi-empirical equations which depend upon experimental evaluation of the numerical constants. No single equation for heat transfer by natural convection from a horizontal surface to the air above has been unanimously accepted. Most of the equations have been reduced to the following simple form:

$$(q_c/A) = K\,\Theta^n \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

where

(q_c/A) = component of total heat emitted by natural convection, Btu per (hour) (square foot).

Θ = temperature difference between the surface and the air, Fahrenheit. The point of measurement of the air temperature has not been clearly defined, but is assumed to be in the turbulent region immediately adjacent to the laminar layer that lies along the surface.

K and n are constants which depend on the following factors: (*a*) position of the surface (horizontal, vertical, etc.); (*b*) direction of heat flow (to or from the surface); (*c*) magnitude of the temperature difference; (*d*) mean temperature; (*e*) point of measurement of the air temperature; (*f*) size of the surface; (*g*) roughness of the surface.

Table 2 lists a large number of values of K and n, as reported by a number of investigators, together with some physical details of the original experimental studies. The objective of this investigation was to determine whether Equation 5 was applicable to a floor surface with a non-uniform temperature and under both steady state and transient heat flow conditions.

The heat-flow by natural convection at the floor surface was calculated by means of each of the following equations:

1. For the section of floor *warmer* than the adjacent air

$$(q_c/A) = 0.81\,\Theta^{1.12} \text{ (Wilkes and Peterson)} \quad \ldots\ldots\ldots \quad (6)$$

$$(q_c/A) = 0.48\,\Theta^{1.25} \text{ (Nusselt, Henky)} \quad \ldots\ldots\ldots \quad (7)$$

$$(q_c/A) = 0.38\,\Theta^{1.25} \text{ (Griffiths and Davis, McAdams)} \quad \ldots\ldots \quad (8)$$

$$(q_c/A) = 0.28\,\Theta^{1.33} \text{ (King)} \quad \ldots\ldots\ldots\ldots \quad (9)$$

2. For the section of floor *colder* than the adjacent air

$$(q_c/A) = 0.38\,\Theta^{1.00} \text{ (Wilkes and Peterson)} \quad \ldots\ldots\ldots \quad (10)$$

$$(q_c/A) = 0.22\,\Theta^{1.25} \text{ (Nusselt, Henky)} \quad \ldots\ldots\ldots \quad (11)$$

$$(q_c/A) = 0.21\,\Theta^{1.25} \text{ (Griffiths and Davis, McAdams)} \quad \ldots\ldots \quad (12)$$

$$(q_c/A) = 0.15\,\Theta^{1.33} \text{ (King)} \quad \ldots\ldots\ldots\ldots \quad (13)$$

In the range of floor surface and room-air temperatures encountered with residential heating systems, the results obtained by the use of the equations of Jakob and Heilman produced essentially the same values as those by the King equation.

The room-air temperatures were measured at seven levels and at seven stations in the rooms. Preliminary observations indicated that the small temperature differences existing between the air and surrounding surfaces produced a negligible error in the temperature observations due to thermal radiation from

the surrounding surfaces to the thermocouples. Only those temperature readings at the 1-in. and 3-in. levels were considered applicable in the convection equations. Such observations, shown in Fig. 7*a* for a cold day, did not show a difference of more than 1 deg for any given station. The values of the temperature difference Θ were obtained by subtracting the room-air temperatures at the 1-in. level from the temperature of the floor surface immediately below.

3. Total Heat Emission: The convection Equations 6 to 13 were used individually in conjunction with the radiation Equation 3 to obtain values of the total heat emission rate (q_t/A) at various distances from the outer wall. These values were plotted as shown by the four broken-line curves in Fig. 7*c*. Similar calculations of the total heat emission rate, utilizing the air temperatures at the

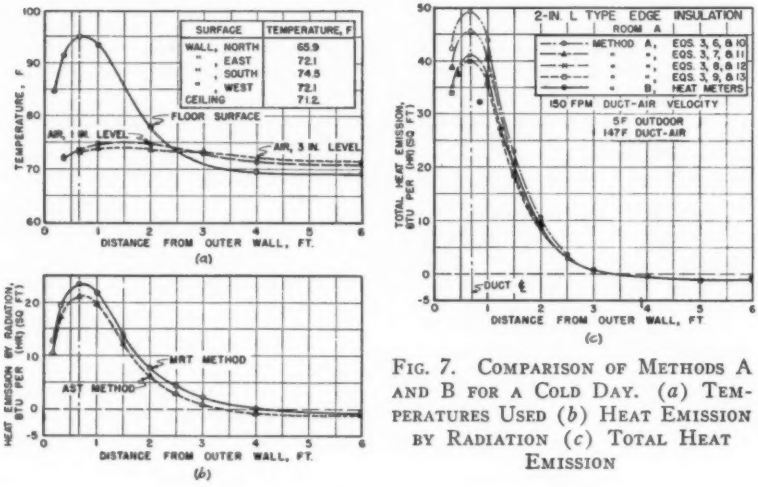

FIG. 7. COMPARISON OF METHODS A AND B FOR A COLD DAY. (*a*) TEMPERATURES USED (*b*) HEAT EMISSION BY RADIATION (*c*) TOTAL HEAT EMISSION

3-in. level for the evaluation Θ in Equations 6 to 13, resulted in an increase of 1.4 percent over the center of the duct and a decrease of 2.0 percent at the center of the room. It can be concluded that for practical applications the air temperature at either the 1-in. or 3-in. levels can be used in evaluating Θ. A comparison of these calculated results by Method A with the measured results of Methods B and C are given in the following sections.

METHOD B—HEAT EMISSION AS MEASURED BY HEAT-FLOW METERS

The eight heat-flow meters in Room A were located on the floor adjacent to the temperature measuring stations and at locations 0.5, 0.83, 1.25, 2, 3, 4, 5, and 6 ft from the outer wall. A confirmation of the meter calibrations, furnished by the manufacturer, was made at the end of the heating season. An additional calibration made on one of the meters by the *National Bureau of Standards* showed an agreement within 3 percent of that supplied by the manufacturer.

TABLE 2—HEAT EMISSION BY NATURAL CONVECTION FROM A HORIZONTAL PLANE FACING UPWARD

ORIGINAL INVESTIGATORS						OTHER SOURCES REFERRING TO SAME DATA						SIZE OF SURFACE USED		RANGE OF SURFACE TEMP USED	RANGE OF TEMP DIFF, θ, USED
Name	Ref No.	Values of K and n[a]				Name	Ref No.	Values of K and n[a]				Length	Width		
		Heat Flow Upward		Heat Flow Downward				Heat Flow Upward		Heat Flow Downward					
		K	n	K	n			K	n	K	n	Ft	Ft	F deg	F deg
Wilkes, and Peterson	10	0.81	1.12	0.38	1.00	—	—	—	—	—	—	8	2	80	5-17
Nusselt and Henky	11	0.48	1.25	0.22	1.25	—	—	—	—	—	—	—	—	—	4-100
Griffiths and Davis	12	0.38	1.25	0.21	1.25	McAdams	9	0.38	1.25	0.21	1.25	3-4	3-4	78-226	25-172
						King	13	0.28	1.33	0.15	1.33				
						Jakob	14	0.275	1.33	0.14	1.33				
						Heilman	15	0.32	1.27	0.16	1.27				

[a] Values of K and n are those in the equation: $(qc/A) = K\theta^n$

The results obtained with the heat-flow meters are shown by the solid-line curve in Fig. 7*c*. Both the calculated and experimental data as represented by the curves indicate an upward heat flow in the peripheral area of the floor and a small downward heat flow in the center area of the room. For distances greater than 2.5 ft from the outer wall the measured heat emission rates by the heat-flow meters were in excellent agreement with the calculated rates based upon radiation Equation 3 and the convection Equations 6 to 13. For distances closer to the outer wall a good agreement was obtained between the measured and calculated values, particularly in the case of the application of the King Equation 9. The total heat emission rates calculated by means of the convection Equations 6 and 7 were only in fair agreement with the measured rates, and were from 12 to 25 percent higher than the values of King. Since the King Equation 9, and to some extent the McAdams Equation 8, showed the best cor-

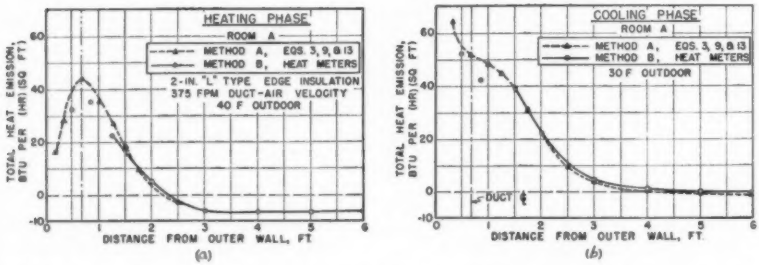

FIG. 8. HEAT EMISSION FROM FLOOR SURFACE UNDER TRANSIENT HEAT FLOW CONDITIONS

(*a*) HEATING PHASE (*b*) COOLING PHASE

relation with the measured results, all subsequent comparisons were made only with Equation 9.

The only noticeable discrepancy between the calculated and measured heat emission curves of Fig. 7*c* was observed in that portion of the curve representing the heat flow from the floor immediately above the perimeter duct. Part of this discrepancy may be attributed to the radial pattern of heat flow from the perimeter duct which produced a lower temperature differential across the hot and cold junctions of the thermopile in the heat-flow meters, and, consequently, a reduced and fictitious value for heat flow.

The data shown in Fig. 7*c* were obtained under steady-state conditions of heat flow. A supplemental study was conducted to determine the rates of heat emission from the floor surface under transient conditions of heat flow. Prior to the start of this study, both the velocity and temperature of the air in the perimeter duct were maintained at constant values. The heat input to the perimeter duct was then doubled, causing the duct-air temperature to rise from 107 F to 157 F in 1½ hr. During this heating phase, the floor surface temperature directly over the perimeter duct increased, attaining a rate of 8 deg per hr approximately 1½ hr after the heat input was increased. The increased heat input was maintained for several days until new steady-state conditions prevailed. The heaters in the electric furnace were then disconnected at the beginning of the

cooling phase, and cold air was circulated through the perimeter duct, thus causing a rapid reduction in duct-air temperature. The cooling rate of the floor surface above the perimeter duct was 14 deg per hr about 1½ hr after the heaters were disconnected.

During this entire study the outdoor temperature remained relatively constant between 30 and 40 F. Measurements of temperature and heat flow were made during both the heating phase and cooling phase approximately 1½ hr following the initial changes in duct-air temperature. The measured rates obtained by heat-flow meters are shown as solid lines in Figs. 8*a* and 8*b*. As in the previous discussion, calculated rates of heat transfer incorporating values from Equations 3, 9, and 13 were determined from the actual instantaneous temperatures. These calculated rates are shown as broken-line curves in Figs. 8*a* and 8*b*. A comparison of the calculated and measured rates shows good agreement between the theoretical and experimental results during these transient conditions.

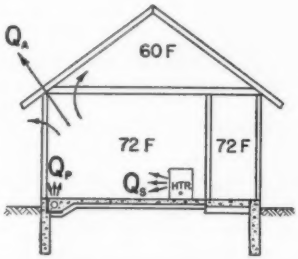

FIG. 9. DESIGNATION OF TERMS USED IN METHOD C

METHOD C—HEAT FLOW AS DETERMINED BY APPLYING A HEAT BALANCE

In both calculated Method A and the experimental Method B, the floor area under consideration was a center strip (only one unit in width) running from the outer wall to a point six feet from the wall. Furthermore, with both methods it was possible to obtain a heat-flow pattern, or gradient, of the types shown in Figs. 7*c*, 8*a*, and 8*b*. The integrated heat flow from this center strip of one unit width is represented by the area under the curves. If the assumption is made that the heat-flow pattern of the center strip is an average pattern for all of the unit-width strips in the room, the integrated heat flow from the entire floor can be obtained by multiplying the integrated heat flow from the center strip by the number of such incremental areas included in the entire floor area.

Since the actual heat losses through the walls and ceiling were offset by the heat gain from the heated floor and that from the supplemental heaters, as shown in Fig. 9, a method was available for evaluating the heat gain from the floor without recourse either to temperature measurements alone (Method A) or to direct measurements by means of heat meters (Method B). Specifically, the

procedure designated as Method C involved a heat balance of the following nature:

$$(Q_p/L) = (Q_a/L) - (Q_s/L) \quad \ldots \ldots \ldots \ldots \quad (14)$$

where

(Q_p/L) = net heat emission from floor, or panel effect, Btu per (hour) (linear foot of exposed edge).
In this study the *net* heat emission represents the heat emitted from a strip of floor one-foot wide and 12 ft in length perpendicular to the outer wall.

(Q_a/L) = above-ground heat losses, Btu per (hour) (linear foot).
The heat losses from the room included transmission losses through the exposed walls and ceiling and the heat required to warm the incoming ventilation air. These were grouped together and have been referred to as the *above-ground heat loss*. This was determined by calibration of each test room. In this procedure, 8 in. of mineral wool insulation was placed on the unheated floors, and daily inputs to the supplemental heaters were recorded. The small amount of heat flow through this insulation was subtracted from the heat input to the supplemental heaters, to obtain Q_a. These values were found to be in close agreement with calculated heat losses.

(Q_s/L) = heat input to supplemental heater during normal operation of the room, Btu per (hour) (linear foot).

The required data included daily records of outdoor, attic, ventilation, and room-air temperatures, and the electrical input to the supplemental heaters. These data were obtained throughout both heating seasons.

The four factors that were shown to affect the temperature distribution of the floor surface were also considered to affect the net heat emission from the floor surface and were therefore investigated in detail as follows:

1. *Duct-Air Temperature:* The curve in Fig. 10*a*, for an average duct-air velocity of 600 fpm, indicated that an increase in perimeter duct-air temperature resulted in a greater net heat emission from the floor surface.

2. *Duct-Air Velocity:* Data are also shown in Fig. 10*a* for duct-air velocities of 150, 375, and 915 fpm. The greater heat emission rates, resulting from the use of higher velocities, were attributed to the reduction in thermal resistance of the inside air film of the perimeter duct. The negative heat emission rates, as shown by the few points in Fig. 10*a*, were obtained with the combination of low duct-air temperatures and velocities. These negative rates occurred when the heat emitted from the section of floor over the perimeter duct was not sufficient to offset the downward heat loss in the remaining area of the floor

3. *Outdoor Temperature:* The scattering of the points representing a fixed velocity and duct-air temperature was partially due to the effect of outdoor air temperature. For example, the group of eleven points indicated by the open circles in the right hand portion of Fig. 10*a* were obtained for a range of outdoor temperatures varying from 15 to 45 F. In order to show the effect of outdoor temperature alone, that is, with constant duct-air temperature and velocity, these eleven points were replotted as shown in Fig. 10*b*. The fact that the heat emission rate increased as the outdoor temperature increased has been attributed to the smaller edge losses and the accompanying higher floor surface temperatures over the perimeter duct. It should be kept in mind that the trend shown in Fig. 10*b* is not directly applicable to the normal operation of a perimeter heating system in a residence since a constant, rather than a variable, duct-air temperature has been maintained.

4. *Edge Insulation:* While the application of insulation at the edge of the slab raised the temperature of the floor at the extreme outer edge a slight amount, and hence resulted in a slight increase in heat emission, the increase was not of great consequence. However, as will be shown in a later report, the edge insulation was effective in reducing the magnitude of heat loss outward through the edge.

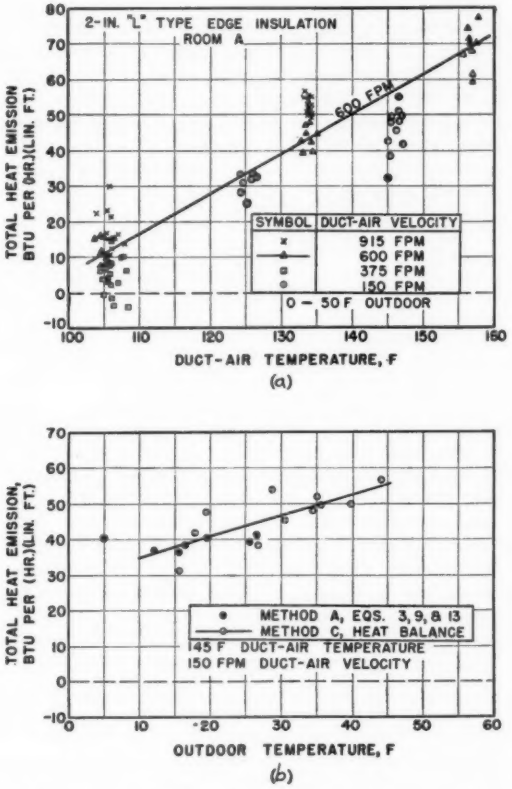

FIG. 10. NET HEAT EMISSION AS AFFECTED BY

(*a*) DUCT-AIR TEMPERATURE (*b*) OUTDOOR TEMPERATURE

The data shown in Fig. 10*b* provides a means for comparing the results by Method C with those by Method A. The solid circles in Fig. 10*b* represent the heat emission rates based on Method A. These rates were determined by integrating the area under curves of the type shown in Fig. 7*c*. The rates thus obtained were found to be in good agreement with those by the heat balance

method. Furthermore, similar comparisons at other duct-air temperatures and velocities showed equally good agreement.

EFFECT OF A FLOOR COVERING

The preceding discussion was based on the heat emission from bare concrete floors. Much speculation has been made as to the effect on the rate of heat emission by the application of a floor covering on a heated floor. In general, the application of any floor covering should reduce the rate of heat emission from the floor due to the additional thermal resistance. However, in the case of rugs or carpets, the possibility has been suggested that the floor covering might increase the heat emission by convection because of the additional surface added by the many fibers in the rug or carpet.

For this phase of the investigation a wool carpet and pad were selected having the following physical properties:

	CARPET	PAD
Material	Woolen felt	Jute
Size, ft	9 x 12	8.75 x 11.75
Weight, lb	26.7	25.0
Thickness, in.	3/8	3/8

The carpet and pad were placed on the floor of Room A with the edge of the carpet flush with the outer wall. The study was conducted during Series A-13a (Table 1) with the duct-air temperature and velocity maintained as in Series A-13, with a bare concrete floor. The heat-flow meters remained on top of the concrete floor and were below the carpet and pad.

Typical heat emission curves for both bare and carpeted floors under similar conditions of operation are shown in Figs. 11*a* and 11*b*. The application of the carpet caused a marked decrease in the rate of heat emission from the heated floor near the duct and little difference in downward heat flow in the center section of the room. The *net rate* of heat emission of the bare concrete floor, as determined from the area under the curves in Fig. 11*a* was 37 Btu per (hr) (linear ft) whereas the rate for the carpeted floor was 21 Btu per (hr) (linear ft). For a constant duct-air velocity and temperature, therefore, the application of a carpet and pad to the floor surface reduced the heat emission approximately 45 percent. While this percentage reduction in net heat emission was large, heat delivery to the room of a residence equipped with a perimeter heating system would not be seriously affected. As was previously mentioned, only a portion of the total heat is emitted through the floor surface and into the room, the remainder being delivered from the registers. As a matter of fact, the system has self-balancing characteristics since any reduction in heat emission from the floor surface would be partly compensated by an increase in register heat delivery.

Comparison of the curves in Fig. 11*a* for the bare concrete floor also shows good arrangement between the experimental results obtained by direct measurements of heat flow (Method B) and those obtained by the application of temperature measurements to the proper heat transfer equation (Method A). In

the studies where the floor was covered with a carpet, as shown in Fig. 11*b*, the calculated results were slightly greater than those obtained by direct measurement. The discrepancy was probably due to the difficulties in obtaining a true temperature of the carpet surface. For this purpose the thermocouples were

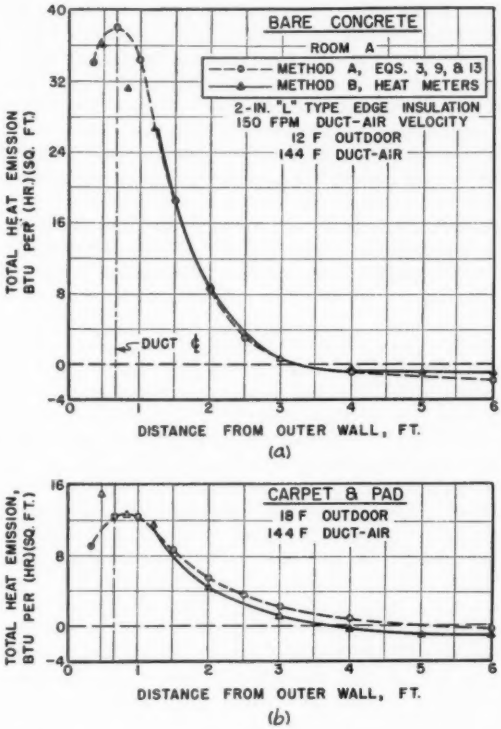

FIG. 11. COMPARISON OF HEAT EMISSION FROM (*a*) BARE CONCRETE FLOOR (*b*) CARPETED FLOOR

taped to the surface of the carpet and probably gave temperature readings that were slightly high[16].

CONCLUSIONS

The perimeter duct installation discussed in this paper consisted of an 8 in. diameter galvanized-iron duct embedded in concrete and located with the center 6 in. below the floor surface and 8 in. from the outer wall. None of the air flowing through the perimeter duct was discharged into the room. The following discussion applies specifically to this arrangement.

The perimeter duct embedded in the concrete floor around the periphery of the building supplied heat to the floor over an area 3 to 4 ft wide. *Increases* in duct-air temperature, duct-air velocity, outdoor temperature, and amount of edge insulation increased both the floor surface temperatures and the amount of heat emitted from that area. One of the limiting features of any floor heating system is the magnitude of the floor surface temperatures experienced. In this connection the A.S.H.V.E. GUIDE[8] in Chapter 23 recommends 120 F as the maximum floor surface temperature for floor areas at the border of a room. Results of this investigation indicated that in order to obtain a floor surface temperature of 120 F an average perimeter duct-air temperature in excess of 180 F would have been required. However, since current practice[17] in warm air heating of residences utilizes a maximum bonnet-air temperature of only 165 F at design conditions, little danger exists of attaining floor surface temperatures as high as 120 F.

Three independent methods for determining the heat emission rates from a floor surface were investigated. In Method A both the radiation and convection components of heat transfer from the floor surface were separately calculated by means of available equations. The radiation component, as defined by the Stefan-Boltzmann equation, was determined by two different procedures. The mean radiant temperature (MRT) method of calculating the radiation heat transfer provided slightly higher results (1.4-2.0 percent) than did the average surface temperature (AST) method. The convection component of heat transfer was determined by means of eight empirical equations, four of which applied to the section of the floor warmer than the room-air, and the remainder of which applied to the section of floor cooler than the room air. The calculated heat flow using the King equation for convection and the Stefan-Boltzmann equation for radiation was found to be in excellent agreement with the experimental determination of heat flow by the use of heat-flow meters (Method B).

The calculated values of Method A using both the King and Stefan-Boltzmann equations were also found to be in good agreement with the total heat emission from the floor surface as determined by a heat balance (Method C). It was concluded that the King equation for natural convection and the Stefan-Boltzmann equation for radiation provided means for predicting the heat emission from a perimeter heated floor from known temperature measurements.

For practical design purposes an average curve has been presented to correlate the *net* heat emission from the floor with the average duct-air temperature. In this study the *net* heat emission represents the heat emitted from a strip of floor 1 ft wide and 12 ft in length perpendicular to the outer wall. For average duct-air temperatures of 120 F, 130 F, and 140 F the corresponding *net* heat emission rates were 28, 39, and 50 Btu per (hr) (linear ft of exposed edge). For rooms other than 12 ft in width, a correction for a downward heat flow must be made. If the room is over 12 ft in width, the correction amounts to -2 Btu per (hr) (ft of width) and for rooms less than 12 ft in width the correction amounts to $+2$ Btu per (hr) (ft of width). The placement of a carpet and pad over the floor resulted in net heat emission rates approximately one-half of those for the bare floor for the same conditions of duct-air temperature and velocity.

The magnitudes of the heat emission from a perimeter duct to the room can be best shown from an example. Consider the case of a 24-ft $\times$ 32-ft residence having a peripheral length of 112 ft. A perimeter heating system for such a residence which maintains an average perimeter duct-air temperature of 140 F

would develop a net heat emission, or panel effect, of about 5600 Btu per hr. This panel effect from the perimeter duct alone would be supplemented by the heat gain from the radial feeder ducts leading from the sub-floor plenum to the perimeter ducts. The magnitude of the panel effect from the feeder ducts would depend on the arrangement and lengths of the feeders. The total heat emission from both the perimeter and radial feeder ducts would represent a substantial portion of the total heat required to offset the heat loss from the structure. The balance of the total requirements would be offset by the heated convection air discharged from the registers. Any reduction in the heat emission from the floor surface by the application of carpeting would be partially offset by an increase in the convection heat delivery through the register.

Also, it is of interest to note that in the example mentioned, a heated floor area 3 ft wide extending around the entire periphery represents 37.5 percent of the total floor area. When consideration is given to the additional floor area heated by the feeder ducts, over one-half of the floor area of the small residence would be serving as a floor panel. Strictly speaking, therefore, the warm-air perimeter heating system can be classified as a convection-panel system.

ACKNOWLEDGMENTS

This paper is a report of an investigation conducted in the Floor Slab Laboratory under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station of the University of Illinois.

This paper includes material from a thesis completed under the direction of Prof. S. Konzo and submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering by C. R. Remke. These results will ultimately comprise part of a bulletin of the University of Illinois Engineering Experiment Station.

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DISCUSSION

R. M. STERN, Seattle, Wash.: I would like to compliment the authors on a very interesting and useful paper.

My office has done about three hundred residence layouts in the past three years. Most of these have been small homes, although not necessarily low cost. Probably two hundred of the jobs have used floor registers in preference to baseboard or high-wall.

In my opinion, the one big deterrent to the acceptance of perimeter heating is the lack of suitable floor registers. There are only two makes of floor registers now available that are at all attractive to floor use.

These several floor registers now manufactured do not completely meet requirements in desirable factors in such a register location. The floor register should not only spread the air up across the window, but also should be adjustable in spread and should have means for tipping the air slightly away from the windows to avoid waving of drapes or fluttering. It is desirable to hold the register as near as possible to the wall of the room, *i.e.*, within 4 in., or 5 or 6 in. where the drapes are voluminous.

The register that lets the air gush up from it directly under the drapes is entirely unsatisfactory. Fin spacing should be quite close to prevent small objects from falling into the duct.

We have gone to the trouble of a double handling proposition on the obtaining of suitable floor registers. A suitable core with very close fin spacing and the necessary deflection features is obtained from a certain manufacturer. We then have a local metal working plant take these cores and put on a suitable border and finish the registers. Thus the owner pays about $5.00 for a grille and $11.00 for a register. Sixty or seventy jobs a year under such handicaps indicates there is a demand for such register use.

Most of our floor register jobs do not go into concrete slab-on-ground homes. About 75 percent have been frame floor jobs.

This may indicate to alert manufacturers the potential demand for a good floor register. I hope some will make it easier to obtain the good registers desired.

M. C. GREINER, Los Angeles, Calif.: I would like to ask Mr. Jamieson about the practical aspects of this type of system. (1) How is the duct constructed; of what material, and how is it sealed at the joints—whether or not there are any unusual problems there. (2) I wonder if some of the work that was mentioned earlier on the slot distribution might not properly tie in with this type of system?

K. F. BALDWIN, JR., Berkeley, Calif.: I wish to add our experiences in this line to those of the previous commentator. One further difficulty which we have had recently was brought to light by an experience where we went into an old residence that had been built for about fifteen years. Over a truckload of dirt was removed from the underfloor ducts that had accumulated in unused portions of the system. In this same place, there was a great deal of difficulty due to this dirt and dust which would flow out and onto the draperies and curtains when the system was started at the beginning of each heating season. This is a problem that may not be obvious at the beginning but which gets worse as time goes on. Just what the answer is to this problem, we do not know.

J. E. MURRAY, San Francisco, Calif.: I wonder what the temperature drop of the heated air is between the furnace and final delivery at the registers; and, have there been readings taken on the lag in warm-up time, taking into consideration periodic heating?

AUTHORS' CLOSURE: There are a number of different types of floor registers available today that were designed for warm-air perimeter heating installations. Such floor registers are usually about 14 in. long and 2 or 4 in. wide with a rather close fret spacing. In some instances these frets are constructed so as to diffuse the air leaving the register. The registers are placed over the perimeter duct, and generally under windows or in other areas where large heat losses occur.

The ducts to be embedded in the concrete slab are usually constructed of 24 or 28 ga. galvanized iron. However, many installations have been made using vitrified clay tile, concrete pipe, or fiber tubes. Information pertaining to the installation of a perimeter heating system covering such items as types of registers, register location, duct location, etc. can be obtained from the *National Warm Air Heating and Air Conditioning Association,* 145 Public Square, Cleveland 14, Ohio in its Manual 4.

The use of the perimeter heating type of floor register has greatly reduced the amount of dirt that usually falls into the registers as these registers are much smaller than those used earlier in the gravity warm-air system. The placement of these registers in areas less frequently traversed together with the use of higher duct-air velocities, and the use of filters in the furnace all tend to reduce the amount of dust that is collected and distributed in the perimeter system.

A study was made at the Floor Slab Laboratory at the University of Illinois on various methods of delivering air into the rooms from the perimeter duct. This study included various sizes and types of floor registers, an inclined-face baseboard register, and several types of continuous-slot baseboard registers, all of which were located on or near the outer walls. The results of this phase of the investigation will be reported in the University of Illinois Engineering Experiment Station Bulletin entitled Heat Emission Characteristics of Warm-Air Perimeter Heating Ducts, currently in preparation.

In answer to Mr. Murray's questions concerning the magnitude of the temperature drop of the heated air between the furnace and the registers as well as the time lag of the system, I would like to point out that these items were studied during the 2-year perimeter heating investigation at the Floor Slab Laboratory. The results will be presented in papers entitled Warm-Air Perimeter Heating—Temperature Drop in Ducts Embedded in Concrete Floors, Part II, and Warm-Air Perimeter Heating—Heat Losses From Floor Slab, Part III, currently in press.

No. 1432

FURNACE THERMAL EFFICIENCY DETERMINED FROM FLUE AND JACKET LOSS

By H. L. McPherson*, K. L. Badger** and W. S. Zawada**,
Cleveland, Ohio

THE THERMAL efficiency of central heating warm-air furnaces now may be determined by a simplified flue and jacket loss procedure that may be readily applied by manufacturers of appliances in their own test work.

Selected from a number of methods investigated by the *AGA* Laboratories at the request of the Subcommittee on Approval Requirements for Central Heating Gas Appliances, the new procedure eliminates the use of an air flow measurement device, thereby reducing considerably the time and effort required for the test without diminishing the precision of the former procedure. The committee has adopted the new procedure for determining the thermal efficiency of all central heating warm air furnaces whether of the forced air or gravity type. It does not, however, apply to floor furnaces for which separate requirements exist.

Previous Test Method

The standard test method for determining the thermal efficiency of gas-fired warm-air furnaces was initially established by the committee in 1927. This procedure required measurement of the air flow rate and of the air temperature rise during its passage through the furnace, to determine the amount of heat transferred to the outlet air stream. The ratio of the heat acquired by the air stream to the heat supplied to the furnace by the gas consumed, was taken as the thermal efficiency of the unit. Flue loss also was measured and the remaining balance of the heat supplied was charged to jacket and unaccounted for losses.

While this method proved adequate for a period of some 18 years, the procedure became increasingly involved as various refinements were adopted. For this reason the study of alternate test methods was made, and the precision of such procedures was carefully compared with that of the established method. The study covered both gravity and forced air furnaces. The major factor complicating the established standard procedure for determination of thermal effi-

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Portland, Ore., July 1951.

A study was made at the Floor Slab Laboratory at the University of Illinois on various methods of delivering air into the rooms from the perimeter duct. This study included various sizes and types of floor registers, an inclined-face baseboard register, and several types of continuous-slot baseboard registers, all of which were located on or near the outer walls. The results of this phase of the investigation will be reported in the University of Illinois Engineering Experiment Station Bulletin entitled Heat Emission Characteristics of Warm-Air Perimeter Heating Ducts, currently in preparation.

In answer to Mr. Murray's questions concerning the magnitude of the temperature drop of the heated air between the furnace and the registers as well as the time lag of the system, I would like to point out that these items were studied during the 2-year perimeter heating investigation at the Floor Slab Laboratory. The results will be presented in papers entitled Warm-Air Perimeter Heating—Temperature Drop in Ducts Embedded in Concrete Floors, Part II, and Warm-Air Perimeter Heating—Heat Losses From Floor Slab, Part III, currently in press.

1432

No. 1432

FURNACE THERMAL EFFICIENCY DETERMINED FROM FLUE AND JACKET LOSS

By H. L. McPherson*, K. L. Badger** and W. S. Zawada**,
Cleveland, Ohio

THE THERMAL efficiency of central heating warm-air furnaces now may be determined by a simplified flue and jacket loss procedure that may be readily applied by manufacturers of appliances in their own test work.

Selected from a number of methods investigated by the *AGA* Laboratories at the request of the Subcommittee on Approval Requirements for Central Heating Gas Appliances, the new procedure eliminates the use of an air flow measurement device, thereby reducing considerably the time and effort required for the test without diminishing the precision of the former procedure. The committee has adopted the new procedure for determining the thermal efficiency of all central heating warm air furnaces whether of the forced air or gravity type. It does not, however, apply to floor furnaces for which separate requirements exist.

Previous Test Method

The standard test method for determining the thermal efficiency of gas-fired warm-air furnaces was initially established by the committee in 1927. This procedure required measurement of the air flow rate and of the air temperature rise during its passage through the furnace, to determine the amount of heat transferred to the outlet air stream. The ratio of the heat acquired by the air stream to the heat supplied to the furnace by the gas consumed, was taken as the thermal efficiency of the unit. Flue loss also was measured and the remaining balance of the heat supplied was charged to jacket and unaccounted for losses.

While this method proved adequate for a period of some 18 years, the procedure became increasingly involved as various refinements were adopted. For this reason the study of alternate test methods was made, and the precision of such procedures was carefully compared with that of the established method. The study covered both gravity and forced air furnaces. The major factor complicating the established standard procedure for determination of thermal effi-

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ciency is the measurement of the air flow through the furnace. Since a precision within one percent is required, the *AGA* Laboratories have been measuring air flow by an electrical method.

The method determines indirectly the rate of air flow passing through it by measurement of independently applied heat necessary to produce a constant temperature rise of the air stream. The heat is supplied electrically and is measured in terms of watts. For accuracy the air temperature cannot exceed the range of

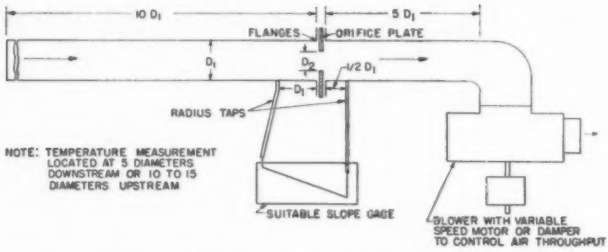


FIG. 1. CONSTRUCTION OF ORIFICE METER

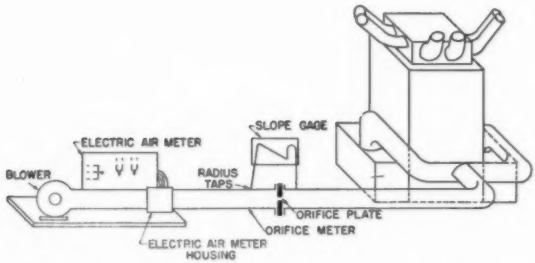


FIG. 2. ARRANGEMENT OF APPARATUS FOR DETERMINING THERMAL EFFICIENCY OF GRAVITY OR FORCED AIR FURNACES WITH AIR FLOW METERS CONNECTED TO FURNACE AIR INLET (GRAVITY TYPE SHOWN)

normal ambient temperatures. Consequently, air flow must be measured as it passes through the instrument prior to entering the furnace inlet. Flow rates are limited to those below 1700 cfm. The design of a meter for employing the electrical method places certain restrictions on the method in which it may be used. While the electrical method is satisfactory for the purpose, constant supervision of the procedure is essential and a period of at least 20 min is required to determine a single flow rate.

An alternate means of determining the amount of heat (or thermal efficiency) delivered at the outlet of a furnace under service conditions is to measure air flow through the outlet connection. In order to do so, however, a flow measuring device capable of operating under a wide range of air flow rates and at temperatures of from 140 to 190 F, is required.

TABLE 1—ADVANTAGES AND DISADVANTAGES OF VARIOUS AIR METERING DEVICES

MEASURING INSTRUMENT	ADVANTAGES	DISADVANTAGES
1. Pitot tube	Precision high. Convenient in measuring velocity at any point within the air stream.	Point measurement. Traverse readings necessary. A velocity head of 0.005 to 0.020 in. of water in the outlet duct would require a micromanometer to indicate differential pressure accurately.
2. Anemometer; swinging vane or rotating vane	Easily operated. Convenient to measure velocity at a point within the air stream.	Point measurement. Limited precision from 5 to 15 percent. Must be calibrated repeatedly.
3. Thermocouple or resistance anemometer	Precision high when properly calibrated. Excellent for very low velocities.	Point measurement. Very small diameter wires required making instrument too delicate for general use in furnace testing.
4. Thin plate square edge orifice plate	(a) Inexpensive and easy to construct (b) Instantaneous accurate reading (c) Gives large pressure differential at low rates of flow (d) Minimum human element involved.	(a) Requires 10 duct diameter upstream and 5 duct diameter downstream (b) Requires blower to overcome resistance of orifice plate.
5. Air flow nozzles	Same advantages as for orifice plate.	More difficult to construct than orifice plate and gives less pressure drop at equivalent flow through orifice of equivalent diameter.
6. Electric air meter	Precision within 1 percent.	(a) Expensive (b) Limited to measurement of air at room temperatures (c) Large and bulky (d) Requires skillful manipulation (e) Limited flow capacity.
7. Venturi meter	(a) Instantaneous accurate reading (b) Low overall pressure drop (c) Wide range of flow and temperature for single meter.	(a) Expensive to construct (b) No interchangeable parts as in orifice meter (c) Less pressure differential than orifice meter.

AIR FLOW MEASUREMENT

In order to assist in the preliminary evaluation of the precision and practicability of various methods for measuring air flow, a literature survey was made. This indicated that an orifice plate or nozzle employed as an air flow measuring device appeared to offer possibilities. Among other metering devices or methods reviewed were Pitot tubes, anemometers of both swing vane and rotating vane types, thermocouple and resistance anemometers, thin plate orifices, nozzle type orifices and venturi meters. Devices which measure air velocity at

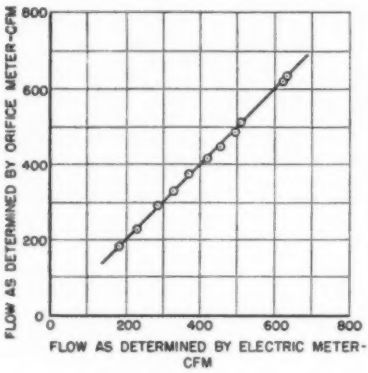


FIG. 3. COMPARISON OF FLOW RATES AS DETERMINED BY ELECTRIC AND ORIFICE METERS

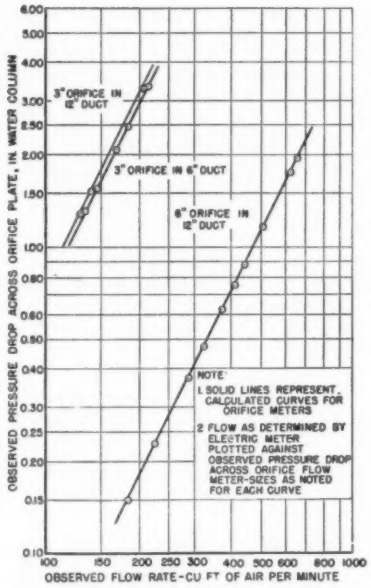


FIG. 4. ELECTRIC METER FLOW *vs.* PRESSURE DROP ACROSS ORIFICE METER

a point, requiring a number of velocity determinations over a given cross-sectional area of duct in order to determine the flow, were discarded since a number of readings must be taken and averaged. Devices of this type include the various anemometers and the Pitot tube. Advantages and disadvantages of the various types of air measuring instruments are shown in Table 1.

Thin plate orifices, nozzles, Venturi meters, and the electrical method indicate air flow in a given duct section with approximately the same precision. Venturi meters, thin plate and nozzle orifice meters have, in general, comparable operating characteristics. The construction of Venturi meters is not as simple as that of orifice meters and requires replacement of the entire meter section when the flow rate exceeds the meter capacity. Orifice meters, on the other hand, may be readily changed from one meter capacity to another by changing orifices.

Thin plate orifices are readily constructed from flat sheet steel, are relatively durable and may easily be replaced. The only machining operation required is to size the orifice and diameter accurately. Aside from these factors, no appreciable advantages are offered by thin plate orifices over nozzle orifices which require accurate machining of a curved surface.

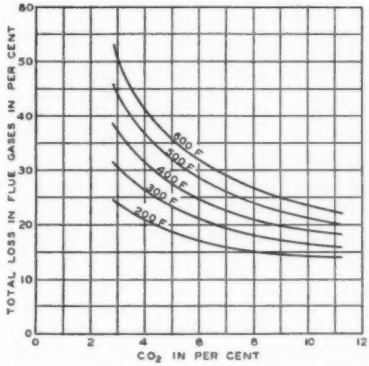


FIG. 5. ALIGNMENT CHART FOR CALCULATION OF FLUE LOSSES FOR NATURAL GAS 1060-1190 BTU

(Curves show flue gas minus room temperature)

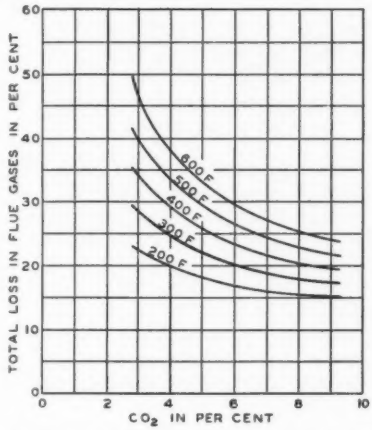


FIG. 6. ALIGNMENT CHART FOR CALCULATION OF FLUE LOSSES FOR COKE OVEN GAS 520-590 BTU

(Curves show flue gas minus room temperature)

COMPARISON OF AIR MEASUREMENT METHODS

In order to determine practical relations between operation of orifice meters and the electric air meter, tests were conducted with each of three sizes of orifice meters, including 3 and 6 in. diameter orifices enclosed in 12 in. diameter ducts and a 3 in. diameter orifice enclosed in a 6 in. diameter duct connected in series with the electric air meter. The orifice meter and the arrangement of test apparatus are shown in Figs. 1 and 2, respectively.

Results of these tests are shown graphically in Figs. 3 and 4. In Fig. 3, rate of flow in cubic feet per minute as determined by the electric method had been plotted against the rate of flow found by the orifice meter. In Fig. 4, for each flow rate, rate of flow as determined by the electric method has been plotted against the observed pressure drop across the orifice under test. Fig. 4 also shows the theoretical curves for flow *vs.* pressure drop based on the equation:

$$Q = k\sqrt{\Delta P}$$

where

Q = flow, cubic feet per minute.
ΔP = pressure drop across the orifice inches water column.
k = constant secured from *Flow Measurements, A.S.M.E.*, 1940.

It is evident from examination of Figs. 3 and 4 that results of orifice and

TABLE 2—COMPARISON OF THERMAL EFFICIENCY TESTS ON *Gravity* WARM AIR GAS FURNACES

APPLIANCE	METER	POSITION	INPUT BTU/HR	HEAT DELIVERED AT DUCT OUTLET %	DUCT LOSS %	THERMAL EFFICIENCY %	FLUE LOSS %	JACKET LOSS %	HEAT BALANCE
A	Electric Meter	Inlet	70,400	68.8	5.2	74.0	20.6	4.9	99.5
	Orifice Meter	Inlet	70,400	69.2	5.2	74.4	20.6	4.9	99.5
	Orifice Meter	Outlet	67,000	67.4	5.6	73.0	21.2	4.9	99.1
B	Electric Meter	Inlet	118,200	65.0	6.2	71.2	26.5	1.0	98.9
	Orifice Meter	Inlet	118,200	64.6	6.2	70.8	26.5	1.0	98.3
	Orifice Meter	Outlet	118,400	64.9	6.0	70.9	26.8	1.2	98.9

electrical determinations virtually coincide, indicating close agreement. In the case of Fig. 3, deviation from a straight line with slope equal to 45 deg would indicate appreciable differences in flow rate as determined by the two methods.

FLUE AND JACKET LOSS DETERMINATION

Approaching the study with the purpose of eliminating the use of a flow metering device, it was decided to compare a flue and jacket loss test procedure with the established test method. Such a procedure could be employed by manufacturers at their plants with a minimum of equipment and also by the *AGA* Laboratories for routine testing. With this procedure, the heat carried away

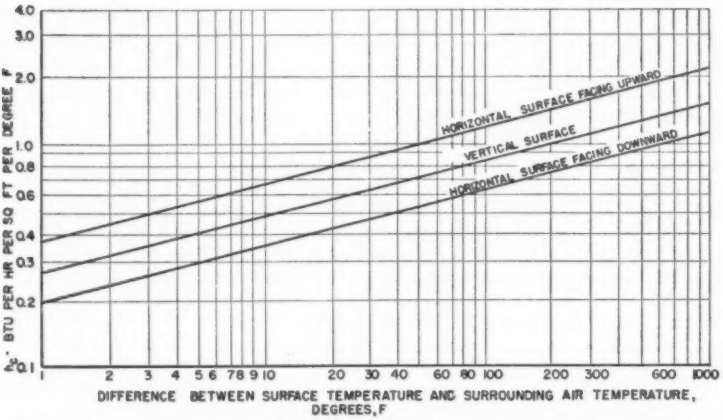
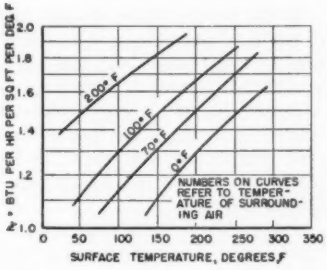


FIG. 7. CONVECTION COEFFICIENTS FOR VERTICAL AND HORIZONTAL SURFACES

FIG. 8. COEFFICIENT OF HEAT TRANSFER BY RADIATION FOR EMISSIVITY OF 1.0



by the flue gases and the heat loss by convection and radiation from the jacket is subtracted from total heat input to the appliance.

The percentage of heat carried away by the flue gases was determined from charts showing flue loss as the function of the temperature and CO_2 content of flue gases. Charts employed when using natural or manufactured gases are shown in Figs. 5 and 6. Jacket temperatures were measured by means of a surface type thermocouple.

The jacket losses of the appliances used during the study were calculated from formulas recorded in the literature for calculating heat transfer from surfaces commonly encountered in central heating appliances, together with convection and radiation coefficients.

For a given jacket area, the Btu heat loss may be expressed as:

$$q = (h_c + h_{rs})\, A_s\, (t_1 - t_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where

q = heat loss per hour, Btu.
h_c = coefficient of convection, Btu per (hour) (square foot) (Fahrenheit degree).
h_{rs} = coefficient of radiation for surfaces in question, Btu per (hour) (square foot) (Fahrenheit degree).
A_s = area of surface considered, square feet.
t_1 = temperature of surface considered, Fahrenheit degrees.
t_2 = temperature of surrounding air, Fahrenheit degrees.

TABLE 3—COMPARISON OF THERMAL EFFICIENCY TESTS ON *Forced* WARM AIR GAS FURNACES

APPLIANCE	METER	POSITION	INPUT BTU/HR	HEAT DELIVERED AT DUCT OUTLET %	DUCT LOSS %	THERMAL EFFICIENCY %	FLUE LOSS %	JACKET LOSS %	HEAT BALANCE
C	Electric Meter	Inlet	95,800	74.1	2.0	76.1	22.8	1.6	100.5
	Orifice Meter	Inlet	95,800	73.6	2.0	75.6	22.8	1.6	100.0
	Orifice Meter	Outlet	96,500	73.0	2.1	75.1	23.2	1.6	99.9
D	Electric Meter	Inlet	84,000	73.0	2.5	75.5	21.0	3.1	99.6
	Orifice Meter	Inlet	84,000	72.8	2.5	75.3	21.0	3.1	99.4
	Orifice Meter	Outlet	83,800	72.6	2.5	75.1	20.7	3.1	98.9

In the beginning an average temperature was established for the entire jacket of the unit undergoing test. This average temperature was then used in Equation 1. An average value of h_{rs} and h_c was also employed to calculate jacket loss in that equation. However, since convection and radiation coefficients vary with temperature, a more precise evaluation of the jacket loss could be made by collecting the total surface areas whose temperatures were approximately the same and determining the h_c and h_{rs} for those surfaces. In this manner a number of Q values would be obtained and their sum would be the total jacket heat loss of the unit.

The exact procedure employed is as follows: The jacket surfaces were subdivided into 6 in. squares and the surface temperature of each square was measured. The area of all squares with temperatures less than 100 F above room temperature, in increments of 10 deg, was totaled. For example, if room temperature was 80 F, areas of squares whose temperatures were between 80-90 F, 90-100 F, 100-110 F, and so on to 180 F were determined. The coefficients were then determined for each group of squares from Figs. 7 and 8. Q was established for each group from Equation 1. This procedure was followed with surfaces whose temperatures were more than 100 deg above room except that areas were collected at increments of 20 deg apart. Thus the total of all the Q values obtained was the jacket and duct loss of the unit.

Since Fig. 8 is based on emissivity of 1.0, coefficients of radiation h_r obtained therefrom must be multiplied by the emissivity of the surface under consideration to obtain the corrected coefficient of radiation h_{rs}. While an average emissivity factor of 0.9 may be used for most jacket surfaces employed with central heating equipment, emissivities for various jacket surfaces may be ascertained from the literature when greater accuracy of results is desired.*

CONCLUSION

For purposes of comparison, three test methods were employed for determining thermal efficiency of gravity furnaces selected during the study. These methods consisted of (1) the present method as outlined in the requirements for central heating gas appliances, wherein the furnace jacket was sealed and air measurement made at the furnace inlet by electrical method, (2) a similar method employing an orifice meter for measuring air, and (3) a method using the flue loss and jacket loss of the furnace. Results are given in Table 2. Likewise, similar tests were conducted on forced warm air furnaces, with results shown in Table 3.

From Tables 2 and 3 it may be noted that if the flue and jacket loss in each test method are added and then subtracted from 100 percent, the resulting thermal efficiency is very close to that obtained by either of the three methods employed to obtain thermal efficiency by evaluating the sensible heat transferred to the air throughout.

Since the flue and jacket loss procedure gave results comparable in accuracy to other methods and had the advantage of materially reducing testing time, equipment and expense, the committee adopted it as a regular test procedure supplanting the old standard method. It is now in use at the *AGA* Laboratories for determining the thermal efficiency of all central heating warm air furnaces of other than floor furnace types.

* *Heat Transmission*, by W. H. McAdams (McGraw-Hill Book Co., Second Edition, 1942).

1433

No. 1433

HEAT PUMP APPLICATION TO A NEWSPAPER PLANT

By J. Donald Kroeker*, John H. Bonebrake**, and James A. Melvin†, Portland, Ore.

THE new *Oregonian* Building (Fig. 1) in Portland, Ore., completed in June 1948, was built specifically as a production plant for a modern newspaper with a daily circulation of 219,000 and a Sunday circulation of 276,000. It contains offices for editorial and business personnel associated with the production plant, and houses a radio station and studios connected with the publishing operation. The building extends five stories above, and two below ground level, and occupies completely a 200 × 200-ft city block. It is a solid block building designed specifically for newspaper production and consequently does not contain ventilating courts. It has a floor area of 220,000 sq ft and a volume of 4,270,000 cu ft. It is of steel and reinforced concrete construction and is faced with sandstone, granite, and fixed, double-pane windows, the exterior panes of which are of heat-absorbing glass.

Building Use Diverse

The east half of the building is devoted largely to the industrial processes of producing the newspaper. It contains the presses, and extensive paper storage area on the lowest floor; a loading dock on an open drive-through court (Fig. 2) on the first intermediate level; a mailing room the full width of the building, directly above the press room; stereotype, linotype and etching departments above the mailing room; and employees' service areas above these mechanical departments. In all except the last mentioned space, no finish is provided on the concrete structure.

In the west half of the building there are shops, mechanical rooms, and an auditorium in the two floors below the street level. The business office, home

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makers' quarters, auditorium, and the bookkeeping and classified advertising departments are located on the first two floors above the street. The third floor contains executive and editorial offices, news rooms, photographic laboratories, and a library. The fourth and fifth floors are occupied by the broadcasting studios (Fig. 3) offices, and broadcasting equipment and mechanical equipment rooms. Except for those below the street floor, these spaces are finished in a

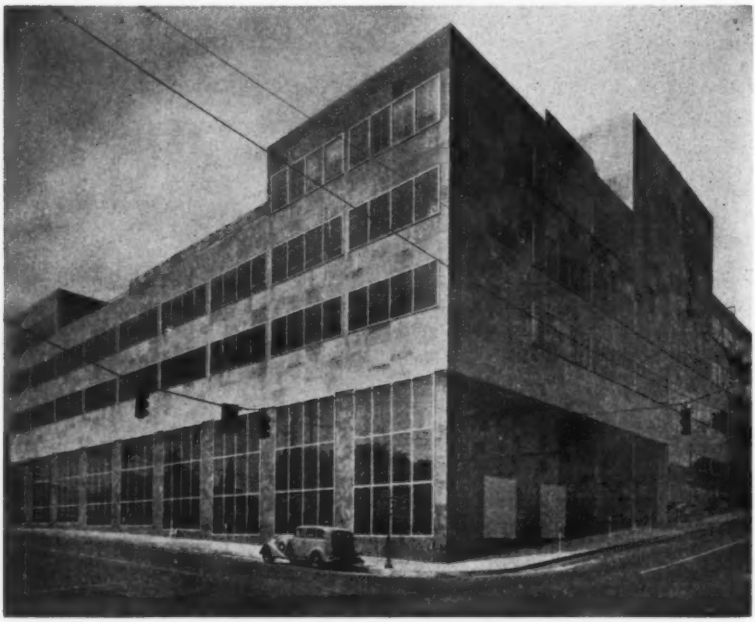

FIG. 1. THE *Oregonian* BUILDING FROM THE PRESSROOM SIDE

conventional manner including acoustical board on metal lath and plaster ceilings.

AIR CONDITIONING NECESSARY

The design of the heating and air conditioning system was based on temperatures of 10 F for heating and 95 F dry bulb with 70 F wet bulb for cooling, the basis used in Portland, and discussed in a previous paper[1]. The average winter temperature of 49 F is significant in studying heat pump system design for maximum practical economy.

While the Portland climate is relatively mild and air conditioning of office buildings is not general practice, it was necessary to use complete air condition-

[1] Exponent numerals refer to References.

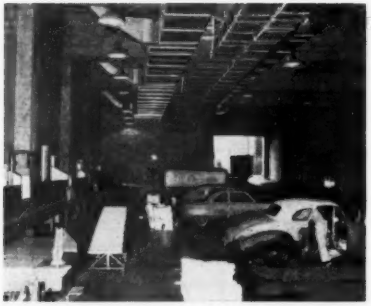

FIG. 2. LOADING DOCKS IN DRIVE-THROUGH PASSAGE

ing in this building because ventilating courts had been eliminated to facilitate uninterrupted flow of material in the mechanical processes of publication. In addition, as the structure is designed for heavy loads, deep steel beams had to be used and these prevented adoption of an all-air heating and air conditioning system.

Heating by air could be properly applied in the press, linotype, stereotype, and mailing rooms. In the west half, largely housing office areas, it was necessary to use a split system, consisting of ceiling panels for base heating and an air distributing system designed for air conditioning and control of space temperatures. A brief description of some of these systems may contribute toward a better understanding of the heat pump application.

FIG. 3. MODERN BROADCASTING STUDIOS ON FOURTH FLOOR

SYSTEM REQUIREMENTS VARIED

Ten air conditioning units supply air at a rate of 204,000 cfm, 99,000 cfm of which is recirculated. Because of the diversity of conditions and the use of various spaces, 90 zones are provided for temperature control. All outside air and recirculated air is passed through electrostatic air filters.

Each of 10 air conditioning units is separately controlled by a program clock, since a newspaper plant operates around the clock and the different departments have varying periods of occupancy and use. Eight of the units are of double-

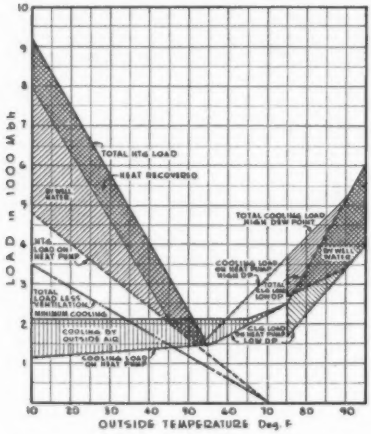

FIG. 4. HEATING AND COOLING LOAD CURVE

plenum type with bottom location of cooling coil and cooling plenum and top location of heating coil. Individual zones are connected by separate ducts to each plenum. The temperature in individual zones is regulated by mixing dampers controlled by room thermostats. Thermostats in all rooms connected to one air conditioning unit are interlocked so that the room requiring the greatest amount of heating will control the warm plenum temperature, and the one needing the most cooling will control the cool plenum temperature.

TABLE 1—BUILDING REQUIREMENTS IN MBH

NATURE	AT 10 F	AT 49 F
Building loss or gain	3500	1250
Ventilation Air	5700	1250
Total	9200	2500

TABLE 2—HEAT BALANCE AT TWO CONDITIONS IN MBH

ITEM	HEATING AT 10 F		HEATING AT 49 F	
	Heat Obtained	Energy Required	Heat Obtained	Energy Required
Heat Pump	4800	1640	1800	360
Well Water Pre-Conditioning	3200	111	—	—
Recovery from Exhaust Air	1200	53	700	70
Totals	9200	1804	2500	430
Less Cooling By-Product		255		310
Net Energy Required		1549		120

In general, humidity control was provided as a high limit only in that no attempt was made to humidify, except in the pressroom. Thus, one humidistat for each unit maintains a maximum of 55 percent relative humidity in the space served by that unit. In the main pressroom where it was desirable to maintain 75 F dry bulb and 55 percent relative humidity, with little variation, full humidity control was necessary.

It will be observed that the several heating and air conditioning systems are conventional in most respects. An exception is the use of coils to recover heat from exhaust air, practicable only with a low temperature medium, such as is available in the use of the heat pump.

After the requirements and necessary heating and conditioning equipment were established as described, the heating and cooling loads were analyzed as shown in Table 1 (Fig. 4).

The cooling demand of 6,000 Mbh was taken as 30 percent under the aggregate computed heat gain, because a considerable diversity in operation of units, although indeterminate, was expected, and because cooling loads are less than computed heat gains.[3]

HEAT PUMP FEASIBILITY ESTABLISHED

While early indications were that an application of a heat pump to the building systems appeared economically practicable, giving promise of providing the lowest available operating costs, the decision to use the heat pump was delayed until a complete analysis could be made. A summary of this analysis is shown in Table 2 and is based on well water availability and temperatures reasonably assured by previous experience with wells in the general area. The analysis

TABLE 3—WELL CHARACTERISTICS

WELL	TOTAL DEPTH (Ft)	PUMPING DATA		TEMP (F deg)
		Flow (Gpm)	Head (Ft)	
DP-1	235	600	145	54
DP-2	204	600	142	55
DP-3	930	500	263	58

indicated that the contemplated application could result in a heat pump load of 4,000 Mbh for cooling and 4,800 Mbh for heating, providing that well water was used for the bulk of the pre-heating and pre-cooling of ventilation air, as well as for series flow through the evaporator and condenser during periods requiring cooling only.

Elements of the heat pump system include drilled wells to provide the well water which serves as the heat source; deep well pumps; a sand-settling, or surge tank for well water; three so-called settling tank pumps to introduce well water into the piping system; two heating pumps; two well-water circulating

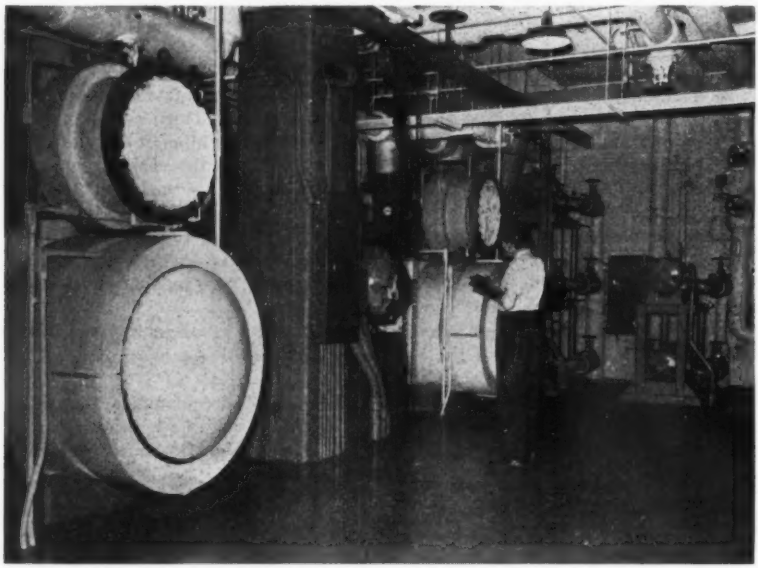

FIG. 5. MECHANICAL ROOM WITH HEAT PUMPS AT LEFT, CIRCULATING PUMPS BEYOND

pumps; two cooling pumps; two refrigerant compressors with heat exchangers (Fig. 5); and various automatic sequence and temperature control devices (Fig. 6).

HEAT SOURCE DEVELOPED

Experience with wells in downtown Portland indicated that water, at about 55 F, should be available in a relatively impervious formation at a depth of 200 ft for use in condensers during cooling. It also indicated that warmer water should be available in a pervious formation at about 500 ft. It was considered that two 200-ft wells at 55 F, and one 500-ft well at 57 F, should be available, the latter for use as a heat sink during cooling. Since it was not possible to

determine drift between the 200-ft wells disposal to sewer was contemplated during use of the deeper well. Actual well conditions developed are shown in Table 3. Wells DP-1 and DP-2 were used to supply water during cooling (with disposal to DP-3). Well DP-3 was used for heating, the water being finally discharged to sewer although this incurred a city sewerage charge.

All pumps used were standard centrifugal units with constant-speed electric motors. They were installed in vertical groups for simplified piping and space saving, with each group having one stand-by pump. The settling tank pumps,

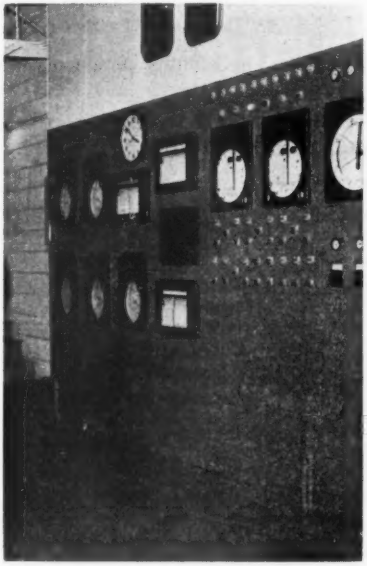

FIG. 6. THE SYSTEM NERVE CENTER IS PANEL BOARD OPPOSITE HEAT PUMP

mounted over the settling or well-water receiving tank, were equipped with automatic primers.

Well water circulating pumps (a relatively minor investment) were provided to precondition ventilating air during both heating and cooling and thereby reduce the load and required capacity of the heat pumps as shown in Table 2.

OPERATION UNDER UNUSUAL CONDITIONS

The refrigerant compressors (using Freon 11) are of the centrifugal true-volute type, with wound rotor motors on 12-step controllers, and suction dampers for operation below surge points. The ancillary heat exchangers are standard shell-and-tube, multi-pass units, designed especially for the application. Nominal

capacity of each assembly is 275 tons. They operate under two unusual conditions:

1. These heat pumps are probably the first of this type that are provided with safe sequences to permit fully automatic operation.

2. Since panel heating was a requisite, and the ceilings in which they were located are finished with acoustical board, a high panel water temperature, or a high condenser water leaving temperature, was required. Consequently, it was necessary to select refrigerant compressors and their heat exchangers for a leaving water temperature of 135 F, higher by 30 deg than such machines in prior normal applications.

FLOW DIAGRAMS TRACED

Control equipment provides, in addition to automatic start, for automatic sequence selection by an outside thermostat and water temperature levels. It

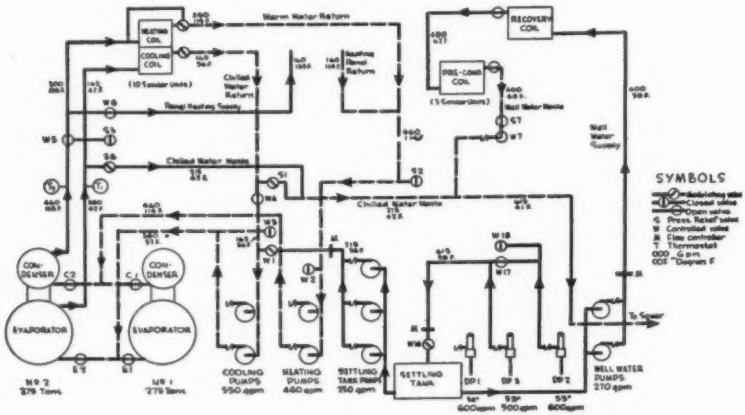

FIG. 7. CYCLE 1, FLOW DIAGRAM, HEATING LOAD THE GREATER

also provides for automatic reset of leaving condenser water temperatures by an outside thermostat.

Operation of the heat pump and the systems served are shown in three flow diagrams. For convenience, they are designated as three cycles specifically:

1. *Cycle 1,* Heating load the greater, Fig. 7.
2. *Cycle 2,* Heating, with cooling the greater, Fig. 8.
3. *Cycle 3,* Cooling above 75 F, Fig. 9.

Actually, as such, there are no cycles. Changeover from one sequence of water flow to another is made automatically as certain control points are reached. This is determined by sequence and heat pump controls mentioned. These diagrams show the significant valves and controls only. Piping and equipment not in use during a cycle are shown unconnected or deleted to simplify study.

Cycle 1: Well water is pumped from DP-3 at 58 F into the settling tank and from there a pump supplies it to the suction line of a cooling pump. This pump

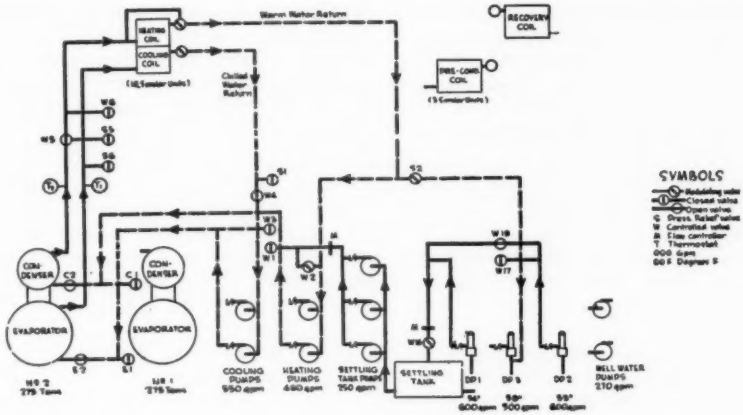

FIG. 8. CYCLE 2, FLOW DIAGRAM, HEATING, WITH COOLING THE GREATER

then supplies it to the evaporators, where a portion of it is discharged directly to the sewer through a pressure relief valve, S6. The remainder is conveyed to cooling coils of the air conditioning systems and is discharged to the sewer through pressure relief valve S1. During this cycle. the heating circuit is closed. In the cycle, water is conveyed by a heating pump to the condensers, where it picks up heat and flows to the heating coils and the heating panels, and finally back to the pump.

A well water pump takes water from the settling tank during the cycle, conveying it through the heat recovery coils in exhaust air streams, where it is heated to 62 or 63 F, then to the pre-conditioning coils and finally through

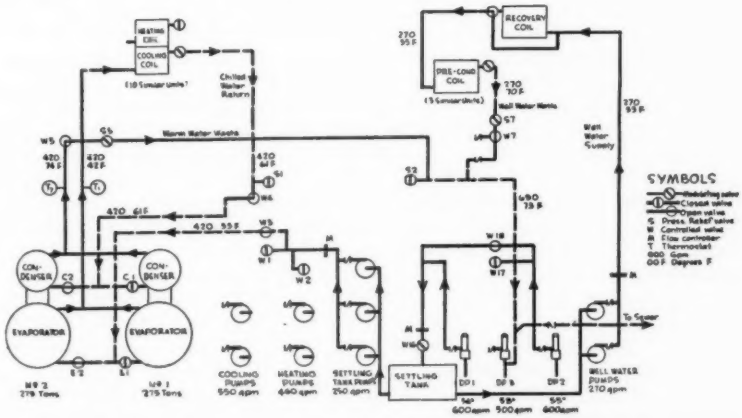

FIG. 9. CYCLE 3, FLOW DIAGRAM, COOLING ABOVE 75 F

pressure relief valve S7 to the sewer. At 10 F outside temperature, the discharge to sewer is at 40 F.

Flow data and progressive water temperatures in each circuit are shown in Fig. 7 for this cycle. When cooling exceeds heating, which occurs when the outside temperature rises to 52 F, valves are automatically repositioned and pumps automatically operated to provide Cycle 2.

Cycle 2: Circuits in Fig. 8 apply during heating above 52 F outside temperature, when the cooling requirement exceeds the heating demands. Well water from DP-1 and DP-2 is supplied to the settling tanks. Pumps introduce 54 or 55 F water to the suction of a heating pump, which conveys the well water to a condenser and open heating circuit. At this time, the panel heating system is off so that the water leaving the condenser flows to the heating coils and from these the excess water is discharged to the casing of DP-3 through relief valve S2.

The cooling circuit is closed during this cycle. Water is circulated by a cooling pump through the evaporator and the cooling coils in a closed system. The well-water recovery and pre-conditioning system is not in operation, because the outside air temperature is too close to the water temperature to permit advantageous operation. When the outside temperature rises to 75 F, and no further heating or heating-up can be required, valves are automatically repositioned and the pumps change to the operation shown to provide Cycle 3.

CIRCUIT REDUCES LOAD

Cycle 3: The heat pump operated for cooling alone is shown in Fig. 9. DP-1 and DP-2 pump the respective wells to the settling tank in this cycle, while the cooling pumps and the heating pumps are idle. The settling tank pumps send well water directly to the evaporators at 55 F, and from these the cooling coils of the several units are supplied. They discharge through the condensers to pick up rejected heat, and then through relief valve S5 to the casing of DP-3 at 73 F at design conditions.

A well water pump also provides 55 F water to precondition ventilation air, by-passing the recovery coils, which provide no advantage, and discharging through relief valve S7 to the casing of DP-3 at about 62 F.

This cycle reduces the load on the heat pumps from the design value of 6,000 Mbh to 4,000 Mbh. Only a part of the reduction is due to the well water preconditioning circuit. The major reduction is accomplished by use of return of water from the coils to the condenser, instead of to the evaporators, thus providing a 6 deg lower temperature of water entering the evaporators.

In all cycles, the start is automatic, and is at the command of the time clock on any of the 10 air conditioning units, with the result that the central equipment operates when any one of the conditioning units starts. Water temperatures are maintained automatically according to outside temperatures and are controlled by means of industrial controllers having thermostatic elements in the discharges of condensers and evaporators, shown as T_2 *and* T_1, respectively, the latter maintaining a constant value of 42 F.

T_1 controls the capacity of the unit or units in operation by increasing or decreasing the speed of the wound rotor motors, which drive the compressors, and also by operation of the suction dampers to the compressors. T_2 in the

condenser discharge, reset by an outside thermostat, controls the amount of well water introduced and determines the circuit into which well water is introduced.

COEFFICIENTS OF PERFORMANCE ANALYZED

With the heat pump system arranged as indicated, coefficients of performance, COP, were computed during design, to determine thermodynamic characteristics and economical efficiencies of the heat pump system and its component parts compared with systems employing conventional heat sources, such as boilers or

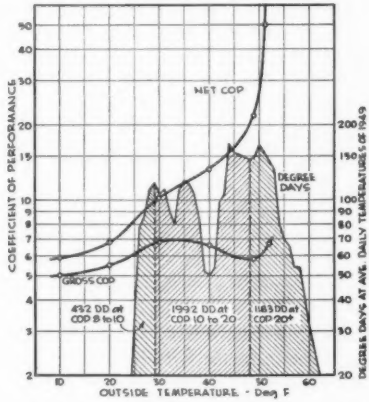
COEFFICIENT OF PERFORMANCE
NET COP
DEGREE DAYS
GROSS COP
432 DD at COP 8 to 10
1992 DD at COP 10 to 20
1183 DD at COP 20+
OUTSIDE TEMPERATURE - Deg F
DEGREE DAYS AT AVG. DAILY TEMPERATURES OF 1949

FIG. 10. GRAPH OF COEFFICIENTS OF PERFORMANCE AND DEGREE DAYS AT AVERAGE DAILY TEMPERATURES OF 1949

district steam mains. Values in Table 2, extended, provided the basis for such analysis.

The *job coefficient* was computed as the ratio of the total heat used in heating the building and in heating the air going into it as related to the total electrical energy used by the system in producing that heat. All units of heat and electricity are expressed in Mbh (thousands of Btu per hour).

The three main sources of heat listed are the heat pump, well water for pre-conditioning, and exhaust air heat recovery. Power to auxiliaries included with power to the heat pump under *energy required* is power for a cooling pump, settling tank pumps, a heating pump, and half of the power for the deep well pump. The other half of the power for the deep well pump, and two-thirds of that for well water pumps, are included under energy required for the well water pre-conditioning system.

The other third of the energy required by the well water pre-conditioning system was charged to the recovery system, since approximately that portion of the energy was required to circulate the water through the recovery coils. In addition, half the energy required for the exhaust fans was charged to the

recovery system, as the coils increased static pressures on the fans by about that amount.

Except as described for the recovery system, energy for the fans is not included under required energy, since they would have required about the same power input, regardless of the heat source. Power to heating and cooling pumps might also be properly exempted, but is included in the interest of conservatism, and to avoid additional complexities involved in considering partial loading. Consequently, the *gross* coefficient of performance was taken as the total heating load divided by the total energy required, as described, and this term is comparable to the usual designation of *heat pump with auxiliaries.*

The *net* coefficient of performance was taken as the gross coefficient of performance with credit for the heating effect, the by-product of the cooling required even at heating design temperatures. The deduction used to arrive at net energy required, Table 2, was the amount of energy required by the compressor for the cooling demand. The *net* coefficient of performance was the ratio of the heating load to the net energy required for the heating at the several outside temperatures studied, including 49 F, the average normal winter temperature.

The coefficients of performance developed are shown in Fig. 10. The net coefficient of performance becomes asymptotic toward infinity at 52 F, inasmuch as heating above this temperature is all a by-product of cooling and is not chargeable to heating.

It will be seen that the high net coefficient of performance for the system as designed (10.2 at 30 F to 20.8 at the average winter temperature of 49 F) and the condition that all heating would be a by-product at temperatures above 50 F, provided an obvious basis for the selection of the heat pump for this application.

Superimposed on the coefficient of performance curves of Fig. 10 is a plot of degree days, computed from frequency of occurrence of average daily temperatures, indicating that the bulk of the heating in 1949 was available at relatively high coefficients of performance. This emphasizes further the significance of the study of average winter temperature data in determining annual costs for energy. Temperatures for a specific year were used here in preference to normal daily temperatures to illustrate a typical year.

INNOVATIONS LISTED

A brief summary of the non-conventional elements of a large heat pump which were applied to this installation includes the following: (*a*) No auxiliary means of heating is provided; (*b*) well water has been used directly to recover heat from exhaust air and to pre-condition ventilation air, both improving the *job coefficient of performance;* (*c*) a completely automatic installation has been provided to reduce the amount of operating personnel and to provide greatest economy; and (*d*) centrifugal refrigerant compressors and heat exchangers have been adapted to provide heating water temperatures as high as 135 F.

OPERATING EXPERIENCE

The *Oregonian* heat pump was placed in operation in June 1948, and was operated manually as the building was occupied before automatic controls were complete. Automatic operation was inaugurated in September 1948, and con-

tinued until December 1948. At that time, due to a personnel problem, the system was again placed under manual operation, and functioned in that manner from January 1949 to May 1949. After the employment of a new chief engineer, however, the system was again placed on automatic control, and has operated automatically ever since.

Conditions of operation were generally in close agreement with the values of flow and temperatures given in the flow diagrams (Figs. 7 to 9). There were some notable exceptions, however. The building was not completely occupied, a large television studio being still unoccupied, and this reduced substantially the actual load on the equipment.

Capacities have been planned to provide for operation of a single heat pump machine down to 25 F outside temperature, the other serving as standby. A single machine has been able to satisfy the heating demands down to 10 F outside.

During design, it was thought desirable to use 54 F water for cooling and 58 F water for heating, with 55 F water added as necessary for either heating or cooling. In practice, it developed that the 55 F water is not required during the heating season, with the result that deep well DP-3 alone carries the heating load.

Experience with other wells below the 200-ft level had indicated presence of a considerable subterranean drift, with the result that, when higher temperature water was delivered to the well, the well temperature on subsequent pumping remained constant at the original value. This was not the case with DP-3, the 930-ft well. At the beginning of the heating season, after warm condenser water has been delivered to this well throughout the summer, water drawn from this well is at 77 F, and, in operation of some six months for heating, it drops only as low as 61 F. This reduces materially the amount of water required from the well.

Another variation found in operation is that the system changes from Cycle 1 to Cycle 2 at temperatures from 2 to 4 deg lower than anticipated. This results in a higher average coefficient of performance than expected, inasmuch as the average winter temperature approximates 49 F.

It has also been found that operation of the heating panels is not required above 52 F, the air systems adequately maintaining temperatures at points above this value. In addition, the starting point of Cycle 3 has become 71 F instead of 75 F as planned.

Despite the fact that development of wells resulted in sandfree water, excessive quantities of sand were settled out in the settling tank and this required considerable maintenance. The sand condition was due to the starting and stopping of the deep well pumps, which shocks the wells. It was overcome by installing a valve in the supply to the settling tank, to modulate the flow and keep the deep well pumps in operation until the required flow from them was approximately 10 percent of the normal capacity. This, too, is accomplished automatically. The modulation of the well water supply to the settling tank has resulted in the collection of less than one cubic yard of sand in nine months of operation.

Another effect of the sand in the system was the scoring of pump shafts which were refinished by stainless steel spraying.

The use of well water in the system has resulted in some corrosion. Steel pipe was used, except in the panel piping, which was copper tubing, and in the coils, which also were copper and contained brass turbolators.

The water in DP-1 and DP-2 contained about 60 ppm of carbon dioxide and was slightly acidic, with iron content substantially below values normally giving concern. The water of DP-3 contained no carbon dioxide and was slightly alkaline, but did, however, contain substantial amounts of chlorides.

Since the system contains a number of circuits in which the flows are variable according to load conditions, it was not practicable to determine requirements for treatment, if any, at the outset. In view of the presence of an acidic water with a high carbon dioxide content, some corrosion was expected and treatment was contemplated, based on the types and amounts of corrosion which might be found in the system soon after it was put into operation. Due to problems with operating personnel, other activities took precedence over early analysis of corrosion conditions. By May 1949, corrosion which might have been arrested earlier became serious and strong dosages of chemicals became necessary. According to corrosion consultants employed, ferrous corrosion carried to the brass and copper in the system, setting the stage for corrosion in these portions also. The system is now being treated regularly with modest amounts of suitable chemicals.

The *Oregonian* heat pump system was not provided with instruments for obtaining detailed records of operating results, such as obtained in the Equitable Building and reported previously[2]. Consequently, it has not been possible to determine the actual coefficients of performance under the conditions of operation found in this installation. Nevertheless, it may be conservatively stated, on the basis of experience with the Equitable Building, and on the basis of the shifting of control points between cycles in the *Oregonian* heat pump system, that coefficients of performance are substantially higher than those shown in Fig. 10.

CONCLUSIONS FROM EXPERIENCE

Numerous conclusions which apply to commercial heat pump installations were drawn from design and operating experience. The following might be useful to others in planning large heat pump installations:

1. The economy in recovering heat from waste products, and the preconditioning of ventilation air with cold water, should be investigated.

2. A commercial heat pump installation can be only as good as the automatic temperature and sequence control provided. Consideration of only industrial types of controls which can obtain greater accuracy and flexibility than usual air conditioning and heating controls is recommended for use in basic master control sequences.

3. Abnormally detailed studies of loads and methods will pay dividends in economy of operation.

4. Determination of economic feasibility requires a rather thorough study of application methods. Comparison of only power rates, arbitrarily chosen coefficients of performance, and costs of alternate energy available, will be misleading and inconclusive.

5. Well water development should precede design, since, in many localities, information based on performance of wells in the vicinity may not apply fully.

6. Sand traps in well water supplies, such as settling tanks, are justified, even for wells apparently entirely free of sand.

7. Water analyses should be carefully studied for possible chemical treatment required to protect system metals. The employment of consultants is advised. Some advice of purveyors of proprietary compounds has been found to be misleading and flavored with prejudice.

8. Shafts of pumps handling well water which show even traces of sand should be made of hardened metals, such as stainless steel, or of the molybdenum group of alloy steels.

9. Generally, a heat pump installation having a capacity of more than 150 tons will require, even though it is fully automatic, at least the part-time attention of an operating engineer, who may also have other duties. Such a person should be personally selected or subject to approval by the designing engineer.

REFERENCES

1. A Heat Pump in an Office Building, by J. Donald Kroeker and Ray C. Chewning (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 221).
2. Heat Pump Results in Equitable Building, by J. Donald Kroeker, Ray C. Chewning, and Charles E. Graham (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 345).
3. Heat Gains are Not Cooling Loads, by C. O. Mackey and N. R. Gay (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 413).

DISCUSSION

LESTER T. AVERY, Cleveland, Ohio: In our part of the country return wells are required for all supply wells. In this paper I note that wells are at two different levels with two different temperatures, the one at the 900 ft depth being 58 F, and the one at the shallower depth being 54 F. I would like to ask whether consideration was given, or could be given, to returning the cold water at one level and the hot water at the other to see if some regain might be obtained. When returning warm water to the ground, we find a gradually increasing supply temperature, which indicates short circuiting. It is suspected that we are running out of cold well water in Ohio.

AUTHORS' CLOSURE (J. D. Kroeker): I'm glad this question came up. As suggested in the paper, wells should be developed before decision is made to use a heat pump system of a water-to-water type.

In a heat pump application which we described in two previous papers, we were able to take the water from one level and dispose of it to another. If we were cooling, we took water from a cooling well, where there was a good drift, and disposed of it into the warmer wells where, unfortunately, there was not much drift, but where there was sufficient capacity to receive most of the water. In this installation we had to dispose of some overflow to the sewer for our heavier cooling loads.

Placing warm water into a well having no drift provides a tremendous thermal advantage as the heating starts.

In the case of the *Oregonian* Building, we encountered well trouble. The first two wells developed were relatively shallow (200 ft deep) and the flow was free. We then indicated that the third well should be drilled to a greater depth, to provide a separation that would permit wasting from one level to another. This well was drilled through solid rock for 500 ft and water was finally reached at 903 ft. In comparable situations in this area, other wells were drilled only to the 500 ft level to reach the lower water.

We expected to obtain 750 gpm with a draw down to about 300 ft. We were able to obtain only 500 gpm. We then decided that it was necessary to make connections to the sewer because we could not discard that water into the upper level (from which

we had to pump for heating) in case we would have the cross circuiting that Mr. Avery has mentioned.

An interesting condition eventually developed in that well. It is now pumping 750 gpm although all we could get originally was 500. In place of a pumping level of 300 ft, we now have a pumping level of about 200. Another interesting point is that the expected drift in this well did not materialize. The result is that when the so-called heating season starts in November, the water supplied by the well is in the neighborhood of 83 to 85 F, much less water is used, and less pumping is necessary.

The warmer temperature in this well at the start of the season reduces to some extent the value of the pre-conditioning circuit for heat recovery and the amount of preconditioning with well water at outside temperatures of 40 to 45 F because recovery cannot be obtained when the water temperature is higher than, or very close to, the temperature of the air from which heat recovery is attempted.

Each well must be treated as an individual case. Analysis of the particular application and use of well water will have to be made after the well is drilled.

The procedure of including wells in the general contract and specifying performance is undesirable. Unfortunately, the architects with whom you work will insist upon specifications for a lump sum figure.

1434

No. 1434

TWO YEAR PERFORMANCE OF A HEAT PUMP SYSTEM FURNISHING YEAR-ROUND AIR CONDITIONING IN MODERN OFFICE BUILDING

By PHILIP SPORN* AND E. R. AMBROSE**
NEW YORK, N. Y.

A 225 HORSEPOWER heat pump system using outdoor air as the heat source and heat sink has been furnishing the year-round air conditioning in the new modern, six story office building of Appalachian Electric Power Co. at Roanoke, Va., for the past two years. Since this system possesses a number of unusual and attractive features, it is believed that a brief description of the design and equipment selection, together with a discussion of performance including an analysis of the electric energy consumed during this period, will be informative and interesting.

THE BUILDING AND EQUIPMENT LOCATION

The building, enclosing a volume of 1.15 million cu ft excluding the garage, is 152 ft long by 96 ft wide by 60 ft high and provides 87,600 sq ft of floor area. The garage has 9150 sq ft of floor area and a volume of 92,000 cu ft. The outdoor air coils are located in the penthouse above the sixth floor and the outdoor air fans are located directly above on the penthouse roof. The fans discharge the outdoor air vertically downward through the coils, then back to the outside. Basement storage tanks charged with warm or cold water, depending on the season, provide practical compressor selections having flexible and efficient operation. All of the other heat pump equipment is located in the penthouse above the sixth floor. Individual room control is obtained by 153 unit room air

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* President, American Gas and Electric Service Corp.

** Air Conditioning Engineer, American Gas and Electric Service Corp. Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

conditioners located throughout the building, and supplemented by a direct system for ventilation air.

A more complete description of the building and of the heat pump system, along with some photographic views of the installation was given in a previous paper.[1]

CAPACITY REQUIREMENTS AND OPERATING CYCLE

The calculated heating and cooling requirements are given in Table 1. The calculated heat loss is 1.41 million Btu per hr at an outdoor design temperature of 15 F, and the calculated heat gain is 2.079 million Btu per hr at an outdoor design temperature of 95 F dry bulb and 78 F wet bulb.

The heat pump consists of three 75 hp, reciprocating, Freon 12 compressors each having its own heat dissipating and heat absorbing surfaces, and so ar-

TABLE 1—HEATING AND COOLING REQUIREMENTS

	HEAT GAIN BTU/HR	HEAT LOSS BTU/HR
Sensible:		
Conduction and Sun Effect — Walls	4,000	
Conduction and Sun Effect — Windows	382,000	674,000
Conduction and Sun Effect — Floors and Roof	62,000	
Lights	531,000	
Ventilation Air	275,000	736,000
People	195,000	
Latent:		
Ventilation Air	480,000	
People	150,000	
Total	2,079,000	1,410,000

Design Conditions
Heating: Indoor temperature ..... 72 F
Outdoor temperature ..... 15 F
Cooling: Indoor temperature 80 F and 50% RH
Outdoor temperature 95 F and 78 F WB

ranged to give three independent, indirect systems. The heating and cooling cycle of one of the three heat pump systems is given by Fig. 1. The circuits of the other two systems are identical, except for the auditorium coil which is in system No. 1 only. This coil furnishes cooling to the auditorium when the main system is on the heating cycle, and provides heating when the main system is on the cooling cycle.

During the heating cycle the path of the refrigerant is from the compressor, through the oil separator, 4-way valve, condenser-cooler, liquid subcooling coil, liquid receiver, expansion valves, outdoor coils, and then back through the 4-way valve to the compressor. The path of the water during this cycle is from the room conditioner units, conditioner coils, booster coils, and/or the storage tanks, through the 2-way valve, condenser—cooler, and the circulating pump back to

[1] Outdoor Air is the Heat Source for Office Building's Heat Pump, by Philip Sporn and E. R. Ambrose (*Heating, Piping & Air Conditioning*, January 1950, p. 138).

the several surfaces. The liquid refrigerant sub-cooling coil, shown in Fig. 1, preheats the ventilation air.

During the cooling or defrosting cycle the refrigerant leaving the compressor flows in turn through the oil separator, 4-way valve, outdoor coils, liquid receiver, expansion valves, and condenser-cooler, and then back through the 4-way

TABLE 2—TYPICAL PERFORMANCE DATA FOR ROANOKE HEAT PUMP SYSTEM WITH OUTDOOR FANS OPERATING AT 565 RPM

	HEATING CYCLE			COOLING CYCLE
	One Compressor	Two Compressors	Three Compressors	One Compressor
Conditioner coil				
Air circuit, cfm	60300	60300	60300	60300
Leaving temp, F	73	74	88	69DB/67WB
Outdoor coil				
Air circuit, cfm	71000	142000	213000	71000
Entering temp, F	78.5	54	44	92
Leaving temp, F	50.0	46	37.5	104.3
Condenser-cooler				
Water circuit, gpm	250	500	750	250
Entering temp, F	93	93	99	67
Leaving temp, F	99	98.5	104.3	61
Electric demand of motors, KW				
Compressor	51.8	101.6	147.9	63.0
Water circulating pumps	4.8	7.8	11.0	4.8
Conditioner fan	25.8	25.8	25.8	25.8
Outdoor fans	21.0	40.8	65.3	21.0
Total	103.4	176.0	250.0	114.6
Compressor head pressure, psig	143	135	149	180
Compressor suction pressure, psig	33	28	20	47
Coil refrigerant effect, million Btu/hr	0.719	1.288	1.564	0.740
Useful heat, million Btu/hr	0.896	1.631	2.068	—
Coefficient of performance based on Electric Energy to compressor and outdoor fan motors	3.6	3.35	2.85	—

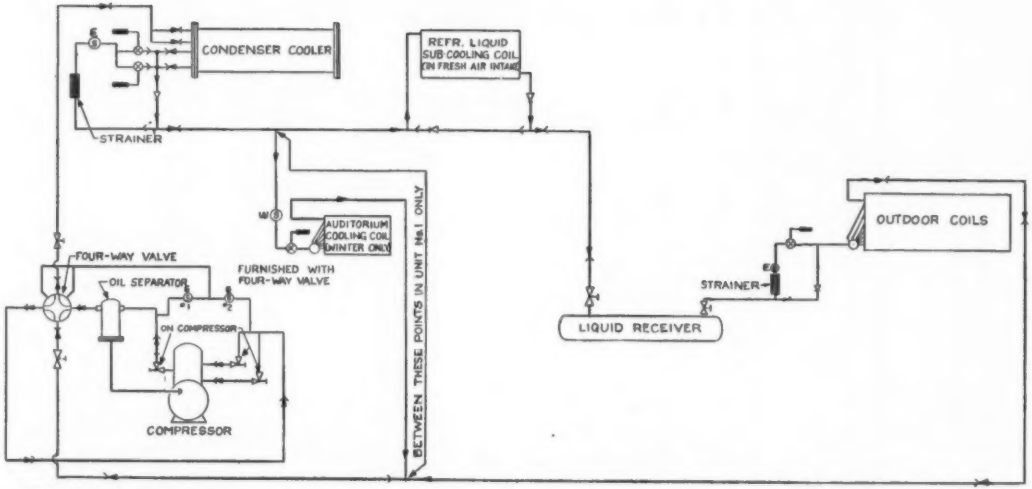
Note:

1. 100 percent of electric energy to compressor motor considered as useful heat.
2. Subcooling effect of the liquid refrigerant in the condenser-cooler and the fresh air coil considered useful heat.
3. This performance data can only be considered as approximately correct due to the difficulties of obtaining reliable air and water temperatures, and air flow measurements in the field. The results are believed to be sufficiently accurate, however, to indicate that the system is performing in accordance with the design requirements.

valve to the compressor to repeat the cycle. The path of the water during the cooling cycle is the same as that for the heating cycle.

Garage heating is independent of the main system. It is accomplished by circulating water from the storage tank through ceiling units by means of a fourth pump. In addition, warm water from the garage system can be circulated through the embedded coils in the concrete ramp to the garage when snow melting and ice prevention are required.

Fig. 2 gives the useful heating capacity of one of the three heat pump units at several outdoor air and suction refrigerant temperatures. The heating effect, due to subcooling the liquid refrigerant in the condenser-cooler and the ventilation air coil, is included. The outdoor air fans are operating at 395 rpm. It



TYPICAL REFRIGERANT FLOW DIAGRAM

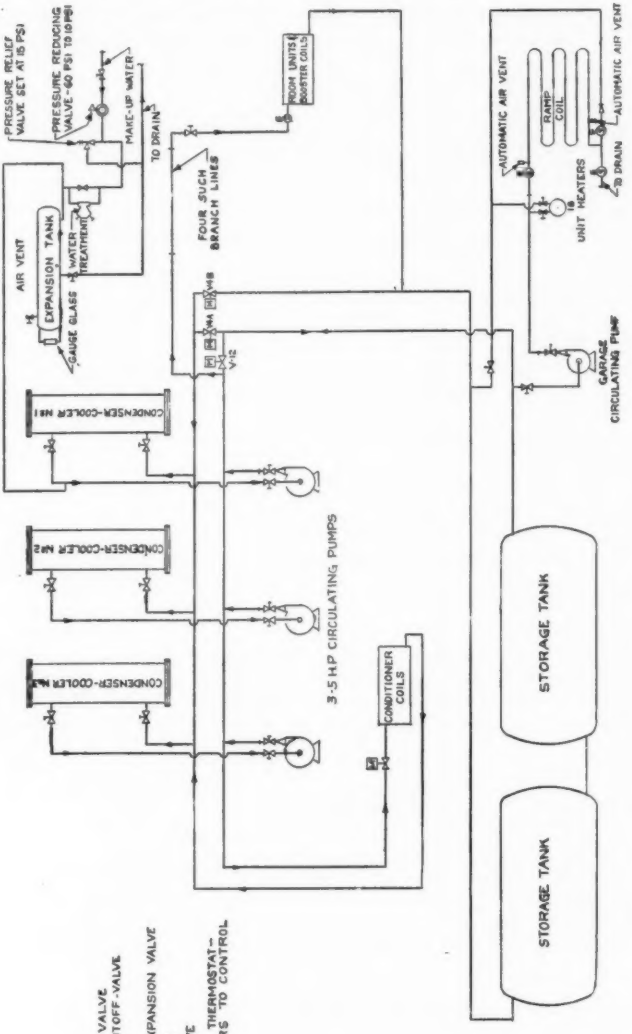
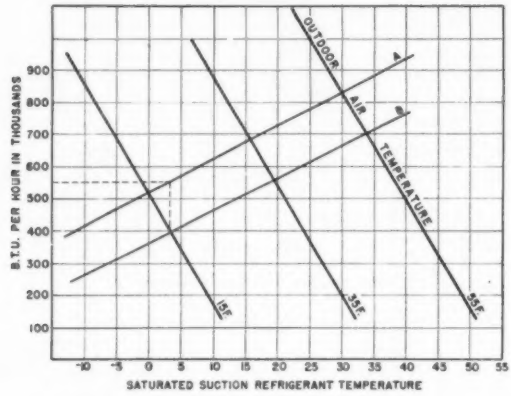


FIG. 1. SCHEMATIC REFRIGERANT AND WATER FLOW DIAGRAM FOR HEAT PUMP SYSTEM



A—Useful heating capacity
B—Outdoor coil refrigeration capacity including refrigerant liquid subcooling effect

FIG. 2. USEFUL HEATING CAPACITY AND OUTDOOR COIL REFRIGERATION CAPACITY OF ONE OF THE HEAT PUMP UNITS AT VARIOUS OUTDOOR AIR AND SUCTION REFRIGERANT TEMPERATURES. OUTDOOR FANS OPERATING AT 395 RPM TO DELIVER 49,000 CFM

can be seen that the heating output of one of the heat pump units is 553,000 Btu per hr at 15 F dry bulb outdoor temperature. The three units, therefore should easily satisfy the building requirements at design conditions. In addition, the storage tanks filled with hot or cold water, depending on the cycle, will supple-

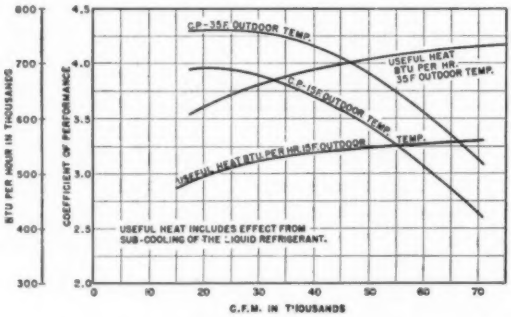


FIG. 3. USEFUL HEAT AND COEFFICIENT OF PERFORMANCE OF ONE OF THE HEAT PUMP SYSTEMS AT VARIOUS AIR FLOWS OVER THE OUTDOOR COILS

ment the compressor capacity. The storage tanks are charged at mild outdoor temperatures when the excess compressor capacity is available.

FIELD TEST DATA

Tables 2 and 3 show a sample of the test data taken during the past two years to check the performance of the Roanoke system. The data given in Table 2

TABLE 3—TYPICAL PERFORMANCE DATA FOR ROANOKE HEAT PUMP SYSTEM WITH OUTDOOR FAN OPERATING AT 395 RPM

	HEATING CYCLE		COOLING CYCLE	
	One Compressor	Two Compressors	One Compressor	Two Compressors
Conditioner coil				
Air circuit, cfm	60300	60300	60300	60300
Leaving temp, F	68	75	67DB/65WB	66DB/64WB
Outdoor coil				
Air circuit, cfm	49000	98000	49000	98000
Entering temp, F	55	33	92DB/74WB	91DB/73WB
Leaving temp, F	42.5	27.0	109.0	107.5
Condenser cooler				
Water circuit, gpm	250	500	250	500
Entering temp, F	92.0	93.0	60.0	59.0
Leaving temp, F	97.8	97.4	54.5	53.5
Electric demand of motor, kw				
Compressor	50.8	97.6	56.0	110.0
Water circulating pump	4.8	7.8	4.8	7.8
Conditioner fan	25.8	25.8	25.8	25.8
Outdoor fan	8.4	16.8	8.4	16.8
Total	89.8	148.0	95.0	105.4
Compressor head pressure, psig	140	135	157	155
Compressor suction pressure, psig	30	18	42	41
Coil refrigerant effect, million Btu/hr	0.694	1.044	0.720	1.400
Useful heat, million Btu/hr	0.867	1.376	—	—
Coefficient of performance based on electric energy to compressor and outdoor fan motors	4.28	3.53	—	—

Note:

1. 100 percent of electric energy to compressor motor considered as useful heat.
2. Subcooling effect of the liquid refrigerant in the condenser-cooler and the fresh air coil considered useful heat.
3. This performance data can only be considered as approximately correct due to the difficulties of obtaining reliable air and water temperatures, and air flow measurements in the field. The results are believed to be sufficiently accurate, however, to indicate that the system is performing in accordance with the design requirements.

are based on the outdoor fans operating at 565 rpm, and Table 3 on the outdoor fans operating at 395 rpm. The fans operated at the higher speed when the system was first installed, but were changed to the lower speed in order to increase the coefficient of performance and at the same time, to reduce the fan noise transmitted to the adjacent building. Operating the outdoor fans at the lower speed raised the coefficient of performance from 16 to 22 percent and reduced the electric demand by 13 to 15 percent. The air and water temperatures, electric demand of the various motors, compressor operating pressure, and

TABLE 4—DESIGN INFORMATION ON THE EQUIPMENT SELECTED FOR THE ROANOKE HEAT PUMP YEAR-ROUND AIR CONDITIONING SYSTEM USING AIR AS HEAT SOURCE

	CONDITIONER SYSTEM	LIQUID REFRIGERANT SUBCOOLING (3 REQUIRED)	OUTDOOR SYSTEM (1 REQUIRED)
Compressors			
Nominal hp	3 - 75		
Speed, rpm	875		
Refrigerant	F-12		
Condenser-cooler			
Nominal tube length x outside diameter, inches	96 x 24		
Passes	8		
Tubes	380		
Tube outside diameter, inches	¾		
Effective area, sq ft	580		
Water flow per unit, gpm	250		
Water circulating pump			
Nominal hp	5		
Speed, rpm	1750		
Capacity, gpm	250		
Total head, ft	50		
Type	Direct Connected		
Fans			
Size	No. 8		No. 8
Type	DIDW		DIDW
Speed, rpm	584		395
Capacity, cfm	60400		49000
Static pressure, inches of water	2.0		0.7
Coils			
Height x nominal tube length, ft	7.31 x 8 and 7.31 x 6	2.44 x 7	8.94 x 9
Face area, sq ft	102.45	17.0	80.45
Total surface: Air side, sq ft	15800	437	12390
Refrig side, sq ft	—	22	634
Water side, sq ft	807	—	—
Depth, rows	6	1	6
Fins, per inch	9	9	9
Tube outside diameter, inches	⅝	¾	—
Air flow, cfm	60400	4000	49000
fpm	590	235	610

the capacity in Btu per hour, during the heating and cooling cycles at the 565 rpm outdoor fan speed as well as the 395 rpm speed, are also given in Tables 2 and 3.

The effect of the outdoor fan speed on the overall performance of the system sometimes is not fully realized. Fig. 3 gives the useful heating effect and coefficient of performance at various air flows over the outdoor coils for this particular installation. The curves show that the air flow resulting in the highest

coefficient of performance for the compressor and outdoor coil combination is about 27,000 cfm. Unfortunately, use of this low air flow seldom results in the most practical installation as far as first cost is concerned. Quite frequently, as was necessary in this installation, some compromises must be made in the design which usually reduces the coefficient of performance below the maximum obtainable.

DESIGN INFORMATION AND EQUIPMENT SELECTION

Some information on design and equipment selected for the heat pump system is given in Table 4. All equipment and the procedures for its selection are the

TABLE 5—DESIGN INFORMATION ON THE ROANOKE HEAT PUMP WATER HEATER

Compressor	
Type	F-12, Hermetic
Nominal hp	3
Condenser	
Surface area, sq ft	135
Cooling Surface	
Height x nominal tube length, ft	2.44 x 3
Face Area, sq ft	7.11
Total surface:	
Air side, sq ft	730
Refrig side, sq ft	37
Depth, rows	4
Fins, per inch	9
Tube outside diameter, inches	¾
Air flow, cfm	4460
Air velocity, fpm	627
Condenser circulating water pump	
Type	Direct Connected
Nominal hp	1/6
Speed, rpm	1725
Capacity, gpm	10
Total head, ft	7

same as those normally used on conventional air conditioning systems. One possible exception is the 4-way valve which was designed especially for heat pump systems. Since some of the equipment is used for both heating and cooling, the equipment must be checked to make certain that the design requirements are fulfilled during both cycles. This applies particularly to the conditioner coil which must be sized to have a definite sensible and latent head capacity during the cooling cycle, and at the same time provide the required sensible heating during the other cycle. The selection given in Table 4 illustrates the types and kind of equipment which have been found satisfactory for this type of application.

DOMESTIC HOT WATER

In addition to the heat pump for year-round comfort air conditioning, a small heat pump supplies the domestic hot water requirements for the entire building. The air exhausted from the building is used as the heat source. This heat pump

was designed to heat 50 gal of water per hr from 50-120 F. The heated water is stored in a 300 gal insulated tank. Exhaust air from the building goes over the compressor and through the cooling coil to the exhaust fan, where it is discharged to the outdoors. The heat removed from the exhaust air by the cooling coil is raised to a higher temperature and transferred, together with the heat equivalent of the work of compression, to the water by means of a refrigerant-water heat exchanger.

Table 5 gives some useful design information and equipment selection for the heat pump water heater. A 3 hp, direct-connected, hermetically-sealed Freon 12 compressor-motor is used. It has been operating satisfactorily since its installation two years ago.

General Performance

The 153 conditioner units and the forced air distribution system which conditions the internal area and supplies all of the ventilation air requirements maintain comfortable conditions throughout the building during both the heating and cooling cycle. On bright sunny days during moderate weather, however, when little heating or cooling is needed, the temperature in the external offices facing west are sometimes several degrees higher than the rest of the building. Although this happens rather infrequently and is not highly objectionable, provisions have been made to modify the electrical controls and piping to zone the conditioner unit on this side of the building.

During one extremely cold and windy period, the water in one of the end turns of the conditioner coil froze, causing a leak. This was probably caused by the ventilation air which passed over the coil before being mixed thoroughly with the recirculated air. Several subsequent cold spells have occurred since then without a recurrence of this trouble.

The heat pump equipment has required relatively little maintenance and service during the two-year period. Two of the compressor shaft seals had to be replaced shortly after putting the system in operation. This trouble was probably due to foreign matter gaining access to the system during the installation. The 4-way valves were erratic in performance and gasket refrigerant leaks developed during the first few months of operation. These difficulties, soon eliminated, are understandable when it is realized that the 4-way valve used on this installation is the first of this size ever made by the manufacturer. The discs of the valves were found to be defective when they were opened recently after almost two years of operation. The manufacturer has since redesigned these discs and it is felt that no further valve trouble will be experienced. The only other operating difficulties were several small refrigerant leaks which were found on four occasions during routine inspection. The outdoor defrost cycle has operated quite satisfactorily from the beginning and no service or maintenance was necessary as a result of frost accumulation.

Electrical Consumption

The monthly electrical consumption of the lights and of the heat pump during the heating and cooling cycle are given in Table 6. The monthly degree days and a calculated heat factor are also included in Table 6.

Attempts have been made to find a constant which could be used to check the periodic operating cost of any particular system, and, at the same time, would

TABLE 6—MONTHLY KILOWATT HOURS FOR LIGHTING AND HEAT PUMP DURING HEATING AND COOLING CYCLE

MONTH	KILOWATT HOURS			DEGREE DAYS	HEAT FACTOR
	Lights	Heating	Cooling		
1949					
April	65,200	28,200	15,000	343	0.472
May	82,800	1,250	49,950	82	0.088
June	64,000	—	54,400	1	—
July	67,200	—	80,000	—	—
Aug	71,200	—	86,000	6	—
Sept	72,000	3,900	55,300	101	0.222
Oct	72,800	12,000	44,000	149	0.463
Nov	75,200	39,600	6,800	536	0.425
Dec	80,800	73,800	1,400	767	0.553
Total	651,200	158,750	392,850	1,985	0.371 (Average)
1950					
Jan	72,800	56,800	4,000	534	0.611
Feb	80,000	65,200	400	686	0.546
March	64,000	73,400	2,600	687	0.614
April	71,200	33,800	16,600	346	0.562
May	67,200	6,200	37,000	55	0.648
June	76,000	200	68,600	10	0.115
July	56,000	—	64,000	—	—
Aug	70,400	—	75,200	—	—
Sept	67,700	1,900	63,700	88	0.124
Oct	66,000	6,100	25,900	167	0.210
Nov	71,200	38,600	11,000	628	0.354
Dec	79,200	111,700	300	911	0.704
Total	841,700	393,900	369,300	4,112	0.449 (Average)
1951					
Jan	68,000	87,700	300	817	0.617
Feb	76,000	98,000	6,000	703	0.801
March	75,600	70,300	2,500	630	0.641
Total	219,600	256,000	8,800	2,150	0.686 (Average)
Two Year Total	1,712,500	808,650	770,950	8,247	0.462 (Average)

Note:

1. Speed of outdoor fans reduced during February 1950
2. Heat Factor=kwhr during the heating cycle/[(kw equivalent of calculated heat loss per degree day) (actual degree days)]

serve as a basis for comparing the operating cost of different heating systems in different geographic areas. One such factor used extensively for this purpose is based on degree days and space volume. The heat factor in Table 6 is based on degree days and the calculated heat loss of the structure. It is equal to the kilowatt hours during the heating cycle, divided by the kilowatt equivalent of the heat loss per degree day and by the actual degree days.

Domestic water consumption and kilowatt hours consumed by the heat pump water heater were recorded for an eighteen month period, but not included in

Table 6. The average daily water consumption was 278 gal, requiring an electrical energy consumption of 38.2 kwhr. Tests showed, however, that the daily tank and pipe loss of the domestic hot water system is 24 kwhr. Consequently, the energy consumption for the useful heat to the water is 14.2 kwhr.

CONCLUSIONS

1. A study of the equipment selection, given as Table 4, not only shows that the same design procedure is used for both the heat pump and conventional air conditioning system, but also that the same equipment except for possibly the 4-way valve, is applicable. The 4-way valve used on this installation and other similar valves are now becoming more readily available. It also might be noted that two 3-way valves may be substituted for the 4-way valve in most of the cycles. This is sometimes done on the smaller installations to reduce the possibility of heat transfer between the high temperature discharge gas and the low temperature suction gas. In the case of the four inch 4-way valves, used on this installation, this heat transfer was found to be practically negligible.

2. Some precaution must be exercised in a heat pump design in order to obtain the most efficient operation within practical limitations. The auxiliaries are the chief offenders in lowering the coefficient of performance. As illustrated by Fig. 3, the outdoor fan alone can easily reduce the coefficient of performance by as much as 16 to 22 percent. Other equipment, such as the circulating pump and conditioner fan, can have an equally adverse effect if not carefully selected.

3. Three separate and distinct systems were purposely used instead of one large system in order to provide more flexible and dependable operation under variable loads. Using three smaller systems instead of one large system also provides means of maintaining a minimum electrical demand. A majority of the commercial electrical rates are based on the maximum demand which occurs during the billing period. An electrical load having a high demand and a low consumption does not favor a low electrical energy cost. By having a multiple number of units, it is possible to decrease the demand by operating fewer units for longer periods. A further advantage of multiple units with a storage system is that possible benefits from off-peak rates may be realized. Loads may be arranged over a 24-hr period in such a manner, with a storage system, that the total maximum demand for all requirements is materially reduced.

4. In general, the heat pump system required relatively little maintenance and service after the first few months of operation. During this initial period, some difficulties were experienced with the 4-way valves, with the compressor shaft seals, and with a few refrigerant leaks. These troubles were soon eliminated, and, with the improved 4-way valve discs, now available, it is believed that the system will continue to perform in a creditable manner.

5. The actual energy consumption during the heating cycle, indicates that the operating cost of a heat pump system, using air as a heat source, is competitive with other fuels in the Roanoke area. The heat factor was computed in Table 6 in order to try to find a constant which can be used to evaluate the heating performance. There is a noticeable difference in the factors for the mild months and for those having severe outdoor temperatures. Part of this difference is undoubtedly due to the wind and sun effect as well as to the heat from the lights which were not considered in the heat factor as computed. Care must be exercised in the use of such a factor until all the influencing items can be properly introduced. It is hoped that with additional data on this and other installations, an acceptable and reliable heat factor can be developed.

6. It is to be noted that the compressor head pressure will be lower with a corresponding reduction in kilowatt hours when water sprays are used on the outdoor

coils during the cooling cycle. Sprays have been used on several such installations but were not believed to be practicable for this system. The installation was so made that the equipment would not have to be relocated or the design changed when future plans for three additional floors materialize. This would have been more difficult to accomplish if evaporative cooling had been incorporated in the design.

7. The heat pump water heater for domestic water is a very practical application which has proven to be very satisfactory during the past two years. It provides maximum efficiency by removing heat from the air being discharged from the building and which otherwise would be wasted. This same type of system has a number of possible applications. The heat removed from the exhaust air can be used to heat the service water, as in the case of Roanoke, or used to heat the ventilation air for the building, as in the case of a hospital where 100 percent outdoor air was required for ventilation. Application of such a design to the latter system improves the efficiency considerably at the lower outdoor temperatures.

8. The heat pump possesses many unique and desirable features which are adaptable to practically any type and design of year-round air conditioning system. The heat pump can provide instantaneous, automatic and simultaneous heating and cooling with the same equipment. A chimney is not required for the unit because no flame of any kind is used and, consequently, there are no products of combustion, soot, dirt or odors. The system requires only an electric utility service and can be located in the basement, on the roof, in the attic, in a utility room, or almost any other place in a structure.

9. It seems reasonable to expect that the number of heat pump installations will increase materially as engineers, architects, and contractors become more familiar with their many desirable features and with the various possible operating cycles. It is hoped that the information given in this paper will serve a useful purpose by indicating the possibilities and potentialities of the heat pump in the air conditioning field and by stimulating and encouraging a greater interest and activity in this type of system.

DISCUSSION

R. C. CHEWNING, Portland, Ore.: What sizes of storage tanks are used? Is it practicable to use tanks of the size required?

J. D. KROEKER, Portland, Ore.: In using the air heat pump in which the refrigeration flow is reversed, I am wondering what consideration has been given to the fact that cooling and heating are required simultaneously in any building of some depth. For example, we have found in water-to-water heat pumps that the load between cooling and heating is roughly balanced by temperatures in one building during outside temperatures of 45 F to 55 F, when no well water is used, and the heat absorbed in the interior of the building is used for heating at the exterior of the building. Has consideration been given to this fact and can it be incorporated in a system in which the flow of refrigeration is reversed, or whether it is necessary that the system be on all heating or on all cooling throughout the building?

G. S. SMITH, Seattle, Washington: Has any trouble been caused by formation of frost on the evaporator?

F. R. ELLENBERGER, Bloomfield, N. J. (WRITTEN): The authors of this paper are to be commended for showing that a large building heat pump system can be practicable, dependable and economical when properly engineered and installed. Their

company has done a great deal of development work in this field, and other organizations would do well to emulate its vision and foresight.

Of particular interest is the heat factor air-conditioner proposed in this paper as a constant for comparing operating cost of systems in different geographical areas. A brief analysis will show that this factor is not actually a constant but varies with the design temperature and as the temperature of the heat source varies—in addition to being affected by variations in wind, sun and lights, as was pointed out in conclusion No. 5. It can be shown that the heat factor proposed is actually equivalent to 2.08/P.F. for a design temperature of 15 F, where P. F. is the performance factor,

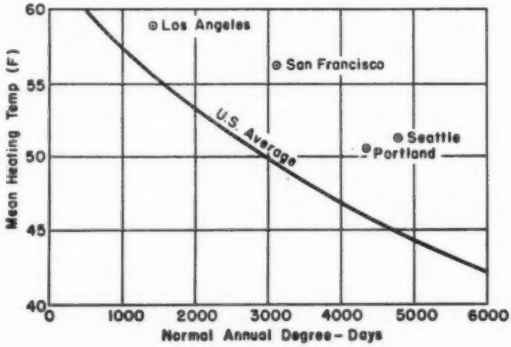


FIG. A. CORRELATION BETWEEN EQUIVALENT HEATING TEMPERATURES AND NORMAL DEGREE DAYS

or coefficient of performance integrated over a particular period.

The discusser recently proposed* a similar heat factor expressed in the form:

$$\text{H.F.} = s\,\frac{\text{Heating KWH}}{\dfrac{\text{(Degree-days)}}{1000} \times \dfrac{\text{(Btu)}}{\text{Hr-°F}}}$$

This form has the advantage of being independent of the heating design temperature and is numerically equal to 7.03/P.F. If the variation of coefficient of performance is essentially linear with outdoor air temperature, as it generally is for an air-source heat pump, then the P. F. can be predicted by taking it as equal to the coefficient of performance at the average outdoor temperature that exists during the time heating is required in any locality. This temperature can be found from weather records and the relationship:

$$\text{Equiv. heating temp} = 65 - \left(\frac{\text{Degree-days}}{\text{No. of days with mean temp. less than 65 F}}\right)$$

As shown on Fig. A from the referenced article*, a surprisingly good correlation has been found to exist in most areas of the U. S. between this temperature and the

* Year-Round Air Conditioning with the Heat Pump—*G. E. Review*, Dec. 1950.

normal degree-days. Most localities, except those noted as separate points, fall very close to the line.

The heat factor mentioned has been used to check a number of instrumented field test installations of packaged heat pumps in diverse geographical locations, and, based upon measured power consumption and measured heat output, has been found to show very good correlation with that predicted by the above method.

It is hoped that the industry will give further study to the use of a heat factor similar to that in the present paper or that described above, in an effort to provide a more useful tool for estimating and comparing operating costs of heat pump installations that would perhaps be more convenient to use than coefficients of performance and performance factors.

It is apparent that the calculated heat loss for the described installation was considerably in excess of the actual heat loss, since the average heat factor of 0.686 is equivalent to a performance factor of 4.5, whereas the coefficient of performance for one compressor operation at 55 F was found to be 4.28, and for compressor operation at 33 F, 3.53. The discusser would be interested in the authors' comments as to the desirability of omitting the power input to circulating pumps and conditioner fans in arriving at the values given, rather than the corresponding figures of 2.82 and 2.73 when they are taken into account. Also, what means were taken to insure that 100 percent of the electric energy to compressor motors is recovered as useful heat?

AUTHORS' CLOSURE (E. R. Ambrose): I am very glad to have Mr. Ellenberger's remarks on the importance of a heat factor in evaluating heat pump performance and I too sincerely hope that more effort will be made in this direction in the future. The close correlation of the heat factor obtained by Mr. Ellenberger on the experimental installations may be partly due to the fact that they were residences void of high internal heat gain. In commercial installations the lighting load and other internal heat gain will materially affect the correlation.

In comparing costs of heat pumps with the more conventional year-round air conditioning systems, it is more convenient to exclude the distribution system, since they are usually the same for both types of systems. In such comparisons the coefficient of performance given in Tables 2 and 3 are generally used. To obtain the total operating costs of the complete system, however, all energy consuming equipment should be included.

No particular means were employed, so that all motor losses would be useful heat. The equipment is located in the penthouse and the heat dissipated is used to maintain comfortable temperature levels in this area.

In regard to Mr. Chewning's questions the two 5000 gal. water tanks are used for the storage of hot or cold water during mild weather for use when extreme temperatures are being experienced. There are four or five such installations on the American Gas and Electric System and we find that they have proved to be practical installations. The one possible disadvantage is the space required by the tank.

In answer to Professor Smith, no trouble has been experienced with defrosting of the outdoor coils. An automatic defrosting cycle is used as described in the reference article which is made a part of this paper.

"The outdoor coils are defrosted automatically during the heating cycle. If the outdoor air-refrigerant suction temperature differential exceeds a predetermined value, indicating accumulation of frost on the coils, the position of the four-way valve is changed so that the high temperature, high pressure refrigerant gas from the compressor goes first to the outdoor coil where it is condensed, giving up its latent heat to defrost the coils. The path of the refrigerant during the defrosting cycle is the same as that for the cooling cycle. The circulating water can continue to furnish heat to the conditioned space during the cycle by using water from the storage tanks."

In reply to Mr. Kroeker, simultaneous heating and cooling of a building is readily adaptable to this heat pump cycle as it is to any other heat pump cycle. In fact,

one of the advantages of a heat pump system is the ease with which such requirements can be incorporated in the design.

The auditorium coil is shown in the flow diagram in Fig. 1. This coil enables cooling to be supplied to the auditorium at the same time that heating is supplied to the remainder of the building. The auditorium is used quite frequently by large groups of people so that cooling is necessary many times during the winter months.

1435

No. 1435

CLIMATOLOGY AS AN AID IN HEAT PUMP DESIGN

By GEORGE S. SMITH*, SEATTLE, WASH.

THE INTRODUCTION of the heat pump in the field of space heating has created a demand for information not generally available. In using any one of the natural heat sources (air, surface or ground water, and the soil) a knowledge of some of the characteristics of the source used is of major importance. A few studies of temperatures and other factors peculiar to the location of the study have been made and others are now in progress.

Eventually the accumulation of information from various parts of the country will permit interpolation of design data sufficiently accurate for most design problems. Fortunately the U. S. Weather Bureau records, which are usually available, will be invaluable as a source of information for any locality under consideration, or to aid in interpolating the results of studies from other localities.

This report contains a considerable quantity of soil temperatures and other data collected in the Pacific Northwest area, and will suggest methods by which Weather Bureau data may be used in heat pump design where the desired type of information is not available and would require much expense and time to obtain.

EXPERIMENTAL APPARATUS USED

Most of the data presented were obtained from three temperature stations. The major station, situated on a golf course at the University of Washington, comprises 13 resistance-type temperature measuring elements spaced from one centimeter to 15 meters below the surface of the soil. Air temperatures are also recorded by similar means from near the surface to a height of 150 ft above the ground. The temperatures are recorded by means of automatic recorders. Recently apparatus to record wind velocity and moisture content of the air, as well as heat flow into the soil, was added, making this the University Clima-

* Professor of Electrical Engineering, University of Washington.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Portland, Ore., July 1951.

tology recording station under the supervision of the Climatology Department. Fig. 1 shows a view of the Climatology Station and Fig. 2 shows the recording equipment. Fig. 3 indicates the positions of the soil thermo elements and heat-flow units.

Two other temperature stations use copper-constantan thermocouples spaced at depths of one, two, three, four, six and ten feet below the surface of the ground. In addition to the enamel and cotton covering, the thermocouple wires are protected by special rubber paint, and a coat of glyptol varnish. They terminate on an enclosed terminal block with metal terminal jacks. These jacks

FIG. 1. UNIVERSITY OF WASHINGTON CLIMATOLOGY RECORDING STATION

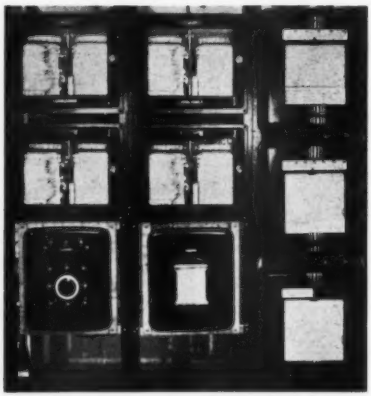


FIG. 2. VIEW OF A PORTION OF THE RECORDING APPARATUS

have tapered holes to match tapered plugs from which the circuit is carried by copper and constantan wires to a second couple on the meter. A portable potentiometer is used to take the readings. The second couple at the meter is taped to the bulb of a mercury thermometer, and the two are carefully encased in a felt covering. The temperature of the couple on the meter is recorded for each thermocouple reading. Readings at these two thermocouple stations are taken weekly.

Weather Bureau records from Seattle, Portland, Spokane, and Boise stations were relied on for much of the remaining information necessary.

TEMPERATURE INFORMATION

The temperatures recorded every 12 min at the Climatology Station (Station A) were averaged for each day. The average air temperatures and soil temperatures at several depths over a period of two years are shown plotted in Fig. 4. A similar plot of maximum and minimum air temperatures as well as

several sub-surface temperatures which were taken at one of the 10-ft depth stations is shown in Fig. 5. This station, designated as Station B, is situated about a quarter mile from Station A, and near the site where a series of tests on ground coils were run. Fig. 6 presents a third set of curves taken at Station C situated in the back yard of a typical town lot about two miles north of the University campus.

There is a striking similarity in the temperatures at the various stations for a given depth. The temperatures at one centimeter depth follow the average air temperature rather closely. However, the continuously recorded temperatures at this depth often indicate a soil temperature several degrees higher than the air temperature a short distance above the surface. An unusually warm or

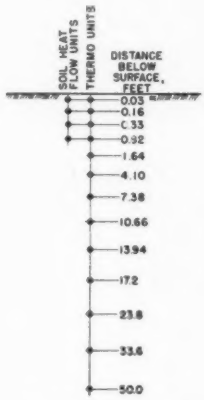


FIG. 3. POSITIONS OF SUBSURFACE TEMPERATURE AND HEAT-FLOW ELEMENTS

cold day will initiate a pronounced temperature wave, conducted rather slowly into the ground, which will appear later at a greater depth with a somewhat reduced maximum. At shallow depths such thermal waves might occur several times a day; at somewhat greater depths they will usually occur daily and at still greater depths less frequently up to the yearly variations. Fig. 7 presents an interesting method of plotting the temperatures recorded at Station A. The lag of the change in temperature at the various depths is clearly shown. There appears to be, in effect, lagging of at least a year even at the 50-ft level, as indicated by the rise of the 50 F temperature line nearly a year after an unusually cool winter. Also the effect of the unusually wet summer of 1948 is indicated by the greater depth of temperature travel.

USE OF TEMPERATURE RECORDS

General Design Factors: Temperature records such as have been described are very useful in heat pump design, especially where ground grids are used as

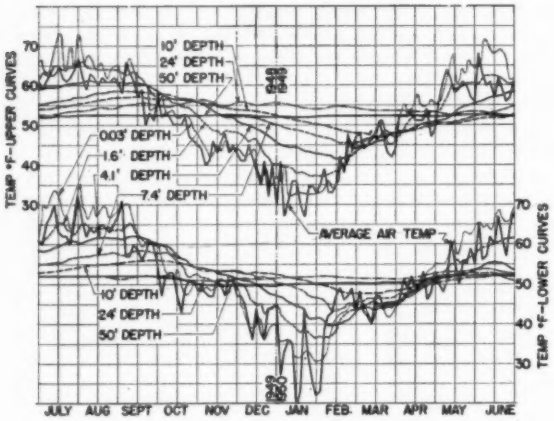


FIG. 4. AVERAGE AIR AND SUBSURFACE TEMPERATURES AT STATION A TAKEN AT THE CLIMATOLOGY RECORDING STATION OVER A PERIOD OF TWO YEARS

heat sources or sinks. The average temperature at the start of the heating season can be found for any depth at which the grid might be placed, assuming that the soil has regained normal temperature conditions during the summer. If the soil has not attained normal temperature conditions the ground grid may not be adequate or it may be of such design that a new lower average soil temperature has been established. In colder climates there is an advantage to be gained by placing the tubes rather deep to gain the advantage of the temperature lag with respect to the atmosphere above the surface. In rather warm climates, and especially where there is considerable sunshine during the heating season, a grid only a foot or two below the surface will gain the advantage of the sun's radiated heat and yet avoid the disadvantage of short cold spells.

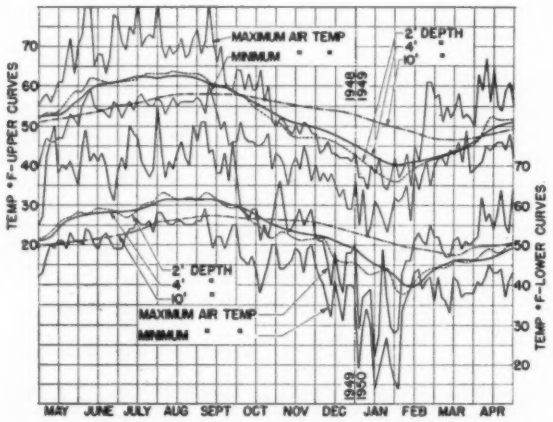


FIG. 5. MAXIMUM AND MINIMUM AIR WITH SUBSURFACE TEMPERATURES AT STATION B TAKEN NEAR GROUND-GRID SITE, OVER A PERIOD OF TWO YEARS

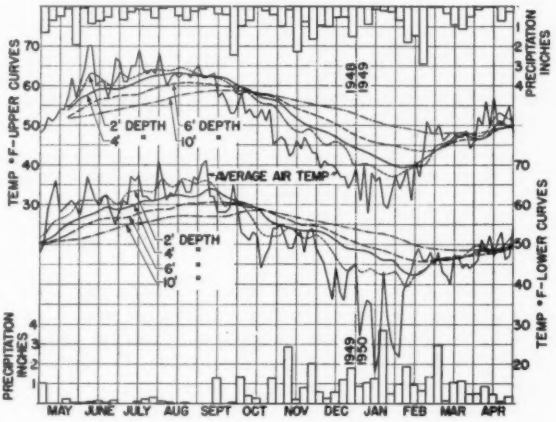


FIG. 6. AVERAGE AIR AND SUBSURFACE TEMPERATURES AT STATION C TAKEN ON A TYPICAL CITY LOT OVER A PERIOD OF TWO YEARS

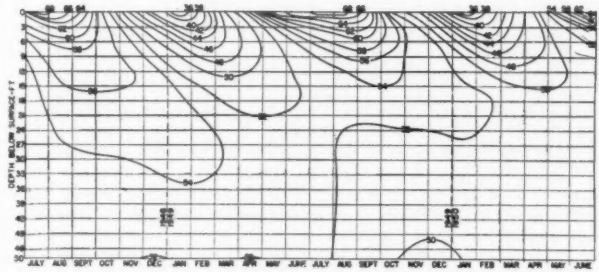


FIG. 7. TEMPERATURE—DEPTH—TIME CURVES—STATION A CONSTANT TEMPERATURE CURVES INDICATE LAG OF TEMPERATURE CHANGES WITH DEPTH. THE EFFECT OF MOISTURE ON THE DEPTH OF HEAT PENETRATION IS ALSO INDICATED

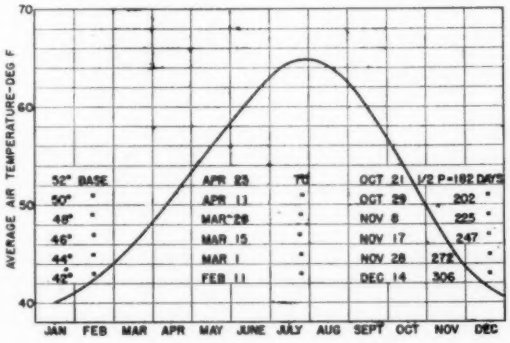


FIG. 8. MONTHLY MEAN AIR TEMPERATURES THROUGH THE YEAR (SEATTLE). THE VARIOUS TEMPERATURE LEVELS ILLUSTRATE THE RELATIVE AMPLITUDES AND PERIODS OF THE POSITIVE HALF OF THE SINE WAVE TEMPERATURE VARIATION

SYMBOLS

A = area — square feet.
a = thermal diffusivity = $k/c\,\rho$ — square feet per hour.
c = specific heat — Btu per (pound) (Fahrenheit degree).
k = thermal conductivity — Btu per (hour) (square foot) (Fahrenheit degree per foot).
Q = heat flow — Btu.
P = time for one cycle of sinusoidal temperature change — hours.
T_o = maximum positive (or negative) value of the sinusoidal temperature change — Fahrenheit degrees.
ρ = density of soil in pounds per cubic feet.
t = time in hours.
x = distance in feet.

Soil Heat Absorption Due to Solar Radiation: The amount of heat available for a ground grid is probably limited to the heat absorbed by the soil from the sun's radiation, plus that precipitated from the atmosphere and carried in by the rain. The solar heat thus supplied to the earth, per square foot of surface may be closely approximated by calculation.

The temperature for the soil at one-centimeter depth, as indicated in Fig. 4, (which will be called the surface temperature) varies rather erratically, but over a year's time it follows through a complete cycle resembling a sine wave with many harmonics. The fundamental sine wave of one complete cycle per year which would produce the same heating or cooling effects in the earth can be closely approximated by the following procedure:

At some depth, usually 30 to 50 ft below the surface of the ground, the soil remains very nearly at a constant temperature equal to the average air temperature above the surface. This constant temperature will be called the base temperature. The difference between the base temperature and the surface temperature will cause heat migration. Over that portion of the year when the surface of the soil is warmer than the base temperature, the average difference in temperature causing heat flow is obtained by dividing the summation of the differences between each day's average surface temperature and the base temperature by the number of days involved. The average amplitude of the positive or negative half of a sine wave is 0.636 of its maximum amplitude. Dividing the computed average difference in temperature by 0.636 will give the maximum amplitude of the equivalent sine wave of temperature changes, which in this case will have a time period of one year.

A solution of Fourier's differential equation[1] for temperature variations of this type may be stated as follows:

$$\frac{Q}{A} = k\,T_o \sqrt{\frac{2\,P}{\pi\alpha}} \qquad \ldots\ldots\ldots\

TABLE 1—SOLAR HEAT FROM SUN RADIATION

YEAR	BASE TEMP START AND FINISH DATES	TEMP DIFFERENCE BETWEEN	TOTAL DEG DAYS DIFFERENCE	½ PERIOD DAYS	AVERAGE DIFFERENCE DEGREES	T_o AVG/0.636	Q/A BTU/SQ FT
1949	42 F						
	Feb 11	Surface and Base	4756	306	15.5	24.4	9550
	Dec 14	71% (Max Air and Base)	4610	306	15.0	23.6	9250
	44 F						
	Mar 1	Surface and Base	4187	272	15.4	24.2	8950
	Nov 28	71% (Max Air and Base)	4070	272	14.9	23.5	8690
	46 F						
	Mar 15	Surface and Base	3604	247	14.5	23.9	8420
	Nov 17	71% (Max Air and Base)	3520	247	14.2	22.4	7900
	48 F						
	Mar 28	Surface and Base	3091	225	13.7	21.6	7260
	Nov 8	71% (Max Air and Base)	3070	225	13.6	21.4	7200
	50 F						
	Apr 11	Surface and Base	2644	201	13.1	20.7	6660
	Oct 29	71% (Max Air and Base)	2560	201	12.7	20.0	6440
	52 F						
	Apr 23	Surface and Base	2258	181	12.4	19.6	5910
	Oct 21	71% (Max Air and Base)	2230	181	12.3	19.4	5860
1950	42 F						
	Feb 11	Surface and Base	4130	306	13.5	21.2	9920
	Dec 14	71% (Max Air and Base)	4100	306	13.4	21.1	9870
	44 F						
	Mar 1	Surface and Base	3553	272	13.1	20.6	9080
	Nov 28	71% (Max Air and Base)	3550	272	13.0	20.5	9030
	46 F						
	Mar 15	Surface and Base	3054	247	12.3	19.4	8140
	Nov 7	71% (Max Air and Base)	3100	247	12.5	19.6	8220
	48 F						
	Mar 28	Surface and Base	2656	225	11.8	18.5	7400
	Nov 8	71% (Max Air and Base)	2720	225	12.1	19.0	7600
	50 F						
	Apr 11	Surface and Base	2310	201	11.4	18.0	6820
	Oct 29	71% (Max Air and Base)	2340	201	11.6	18.3	6920
	52 F						
	Apr 23	Surface and Base	1904	181	10.5	16.5	5900
	Oct 21	71% (Max Air and Base)	1990	181	11.0	17.3	6190

Soil constants used for 1949:
Moisture = 10%, thermal diffusivity $\alpha = 0.018$, thermal conductivity $k = 0.544$, soil dry density = 93 lbs per cu ft
Soil constants used for 1950:
Moisture = 15%, $\alpha = 0.0175$, $k = 0.65$, soil dry density = 93 lbs per cu ft

grid, then the earth will more readily absorb radiation from the sun and will do so over a longer period of time. In order that the maximum ordinate as well as the period of the new equivalent sine wave value may be determined, the following procedure may be used.

Fig. 8 shows a plot of the average air temperatures for each month from the Seattle Weather Bureau's 1900 to 1950 records. Using 52 F, which is Seattle's average yearly temperature, as the zero for the resulting sine wave, the heating and cooling period are almost exactly six months each. If, because of a ground grid the average ground temperature was lowered, say, to 50 F, then the heating portion would have a longer

period, from April 13 to October 28. The maximum value of the equivalent sine wave for the heating portion of the cycle would be found as previously specified using 50 instead of 52 F as the mean yearly average and between the dates indicated. The total period or duration of the cycle would of course be twice the value found for the positive half and thus more than one year in length since the presence of the ground grid has altered the normal cycle. Several other base temperatures are also indicated on the diagram.

The soil-surface temperatures are not normally available and determining them would require time and expense. There should, however, be some fairly

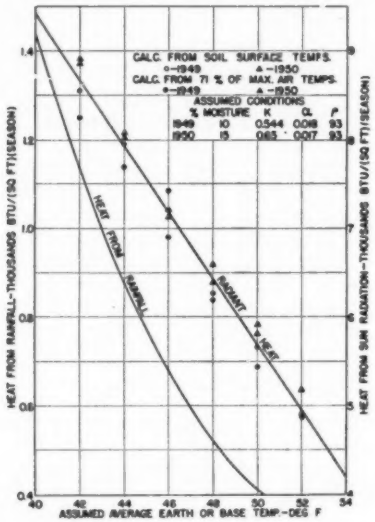


FIG. 9. PLOT OF SOIL HEAT ABSORPTION FROM THE SUN'S RADIATION AND FROM RAINFALL

definite relation between the maximum daily air temperatures, and the average daily surface soil temperature for that particular day. This substitution will of course give only an approximation of the true value, but will in general give results accurate enough for heat pump design purposes. Such information is readily available at any Weather Bureau station.

When temperature differences were calculated by using both maximum air and average surface values against the base temperature, the average ratio of the differences was found to be 0.71 to 1. Thus when using maximum air temperatures, the differences must be multiplied by 0.71.

Table 1 shows the actual calculated heat absorption from recorded surface temperatures as well as those estimated using the maximum daily temperatures with the proper multiplying factors. Fig. 9 shows a plot of these data.

Soil Heat Absorption Due to Rainfall: In many localities, especially in the Pacific Northwest the rainfall probably carries a considerable amount of heat

into the soil and the ground moisture due to rain will greatly increase the soil's capacity to maintain this heat energy in storage. Any accurate method for determining the amount of heat energy derived from this source would require a great deal of study and time. Table 2 presents a method of approximating rainfall heat additions by using the Seattle Weather Bureau's 50-year averages of rainfall for each month together with the average temperature for the corresponding month.

The amount of heat added by rainfall can be readily computed by using as data the weight of rainwater falling on each square foot, and the difference between the average air temperatures and the assumed average or base ground temperature. This method assumes that the temperature of the rainwater is at the average air temperature. Such an assumption is probably quite in error

TABLE 2—CALCULATED RAINFALL HEAT ABSORBED BY SOIL

MONTH	RAINFALL INCHES	LB WATER PER SQ FT	AVERAGE AIR TEMP	BTU ADDED PER SQUARE FOOT BASED ON AVERAGE SOIL TEMP OF				
				40 F	44 F	46 F	48 F	50 F
Jan	4.94	25.7	40.8	20				
Feb	3.89	20.2	42.8	56				
Mar	3.05	15.8	46.4	100	38	6		
Apr	2.38	12.4	51.1	140	88	63	38	14
May	1.87	9.7	56.6	160	122	103	83	65
June	1.33	6.9	61.4	150	120	106	93	79
July	0.63	3.3	65.5	84	71	64	58	51
Aug	0.70	3.6	65.5	92	74	70	63	56
Sept	1.77	9.2	60.3	187	150	131	113	95
Oct	2.84	14.8	53.7	200	148	114	84	55
Nov	5.03	26.2	46.8	178	73	21		
Dec	5.60	29.1	42.6	75				
TOTALS	34			1442	884	678	532	415

during the winter months because a rainy spell will raise the air temperature considerably above the average for the month, while a cold dry spell will generally lower it. Fortunately such an error will result in more heat addition than the calculations indicate. The computed heat values in Table 2 are plotted in Fig. 9.

Validity and Use of Calculated Results: The total calculated heat added per year per square-foot area of soil for any assumed average ground temperature will then be the sum due to the sun's radiation plus that added by the rainfall.

As crude as this method of calculating the yearly heat available in the soil for heat pump use may appear, it does offer some guidance until experience may prove the method too pessimistic. In fact, under the conditions assumed, the soil beneath each square foot of surface will hold little more heat than is indicated by the curves in Fig. 9.

As an example, assume that a ground grid causes the average of the soil temperatures from the surface to a depth well beyond which the grid will draw heat, to drop to 48 F. From Fig. 9 the solar radiation would have supplied approximately 7450 Btu, and rainfall would have added 530, or a total of 7970 Btu

per sq ft of soil area. From Figs. 4, 5 or 6, the average soil temperature over the same range of depth would be about 55 F at the start of the heating season, about October 1. This would result in a 7-deg average temperature differential to promote heat movement. Assuming soil constants found at the location of these tests, we would have the dry soil density as 93 lb per cu ft, and a moisture content at this time of about 15 percent. If the specific heat of the dry soil is 0.2 and for the water contained is unity, then the heat in one cubic foot of dry soil is $93 \times 7 \times 0.2 = 130.2$ Btu while that in the moisture content will be

TABLE 3—PERCENT OF HEAT PUMP OPERATING TIME SEATTLE, WASH.

MONTH	DEGREE DAYS	PERCENT OF TIME OPERATING[a]	HOURS OF OPERATION	PERCENT OF TIME OPERATING			
				House A	House B	House C	House D
Jan.	763	70	520	29		61	93
Feb.	635	65	437	31	54	50	69
March	590	54	400	22	35	33	61
Apr.	434	41	295	19	22	31	38
May	289	26	193	10	9	9	30
June	157	15	108	3	7	4	14
July	65	6	45	0.6	3	1	3
Aug.	68	6	45	1	3	1	28
Sept.	169	16	115	3	5	8	30
Oct.	365	33	245	11	27		
Nov.	556	51	367	22	40	43	42
Dec.	704	65	445	27	39	61	81
TOTALS...	4793		3215				

DETAILED INFORMATION ON HOUSES

HOUSE	GROSS HEATED CU FT	UNIT	HEAT SOURCE
A	10904	3 hp	Well
B	39000	2-5 hp	G. Grid
C	13000	3 hp	Lake
D	33500	5 hp	Spring

[a] Based on average coldest January day for 10 years taken every other year. Degree-Day Average coldest Day=35

$93 \times 0.15 \times 7 = 97.6$ or a total of 227.8 Btu. Dividing the total absorbed heat per square foot of surface by the heat contained per cubic foot will give a required column of soil 35 ft below the surface. At this depth the normal soil temperature changes but little throughout the year, indicating that the normal heat travel below this depth is negligible.

The total annual heat available from an area below which a ground grid is operating is probably somewhat greater than that indicated by Fig. 9. Since the ground grids are placed comparatively near the surface, the temperatures within a considerable distance from the grid will, in general, be far below the average soil temperatures especially after the major heating season. As a result of this abnormally low temperature the portion of the ground near the grid

will accelerate the heat flow from all directions and thus from the atmosphere above when warmer than the soil. Thus heat in excess of that calculated will be absorbed in such an area in the early autumn, in winter if sunshine is available and no snow is present, but especially in the spring and early summer. For this reason the amount of heat as determined by Fig. 9 could safely be assumed as available during the major heating season only, which for Seattle would be from about October 15 to about March 1. For any other locality the major heating season must be determined by a study of the Weather Bureau data.

TABLE 4—PERCENT OF HEAT PUMP OPERATING TIME PORTLAND, ORE.

MONTH	DEGREE DAYS	PERCENT OF TIME OPERATING[a]	HOURS OF OPERATION	PERCENT OF TIME OPERATING	
				House E	House F
Jan	781	68	505	61	59
Feb	613	59	396	54	52
Mar	536	47	349	45	35
Apr	366	33	237	25	24
May	229	20	149	13	
June	105	9	64	7	
July	26	2	15	8	
Aug	28	2	15	12	
Sept	105	9	64	15	
Oct	293	25	186	20	
Nov	537	48	346	38	
Dec	721	63	470	46	58
TOTALS	4340		2796		

DETAILED INFORMATION ON HOUSES

HOUSE	GROSS HEATED CU FT	UNIT	HEAT SOURCE
E	10300	3 hp	Well
F	23000	5 hp	Well

[a] Based on average of the coldest January day for 7 years, taken every third year. Degree days for average coldest day=37

Very little heat from the interior of the earth is available since the earth's natural temperature gradient is about 1 deg increase for every 60 ft increase in depth below the point at which the temperature remains virtually constant throughout the year. Such a small differential temperature will cause very little heat migration.

Underground moisture migration, especially where the surface has a more or less gradual slope, may supply a given length of buried tubing with far more heat than would otherwise be available. Two rather unusual installations of this type have been under observation for some time in connection with these studies. However, with other heat pumps in the near vicinity, such moisture migration might become a detriment.

Determining the Operating Time: Weather Bureau records offer a rather simple means of calculating the approximate percentage of the total time the

heat pump will operate each month. A properly designed heating system should supply the house with adequate heat on the average coldest day that might be expected. In most northern climates such days occur in January. This average coldest day should be chosen by selecting the coldest January day for each of a number of years taken more or less at random, such as every one for five or more years, or every other one for ten or more years. Since the heating system must run almost continuously on this day, the degree days for this average coldest day multiplied by the days of the month considered would give the

TABLE 5—PERCENT OF HEAT PUMP OPERATING TIME SPOKANE, WASH.

MONTH	DEGREE DAYS	PERCENT OF TIME OPERATING[a]	HOURS OF OPERATION	PERCENT OF TIME OPERATING	
				House G	House H
Jan	1152	73	542	74	84
Feb	929	65	437	64	70
Mar	757	48	356	47	48
Apr	492	32	230	27	30
May	281	17	126	13	13
June	114	7	50	8	12
July	21	1	7	9	5.5
Aug	37	2	14	11	4
Sept	187	12	86	9	18
Oct	482	31	230	12	33
Nov	821	53	380	43	62
Dec	1061	67	498	45	73
TOTALS	6334		2956		

DETAILED INFORMATION ON HOUSES

HOUSE	GROSS HEATED CU FT	UNIT	HEAT SOURCE
G	17660	5 hp	Ground Grid
H	10180	3 hp	City Water

[a] Based on average of the coldest January day for seven years, taken every third year. Degree days for average coldest day=51

equivalent monthly degree days. The Weather Bureau's average degree days for a given month, divided by this equivalent value, and then multiplied by 100 will give the percentage of time the heating system will operate during that month. Tables 3 to 6 present such computed percentages using the Weather Bureau's 50-year monthly average degree days, and seven random years to determine the average coldest day. In each case the actual percentage of operating time as computed for houses using heat pumps is also given. In most cases the actual operating time is less than that computed. This either indicates that the unit is larger than necessary or that heating is being conserved, as is the case when the house is not heated at night, or unoccupied temporarily. In other cases the actual operating time is greater than the computed time which might indicate an under-sized heating unit.

The computed operating time for the various climates result in some interesting differences. Thus, the operating times in Seattle or Portland are rather high and spread throughout most of the year though the degree days per year for Spokane or Boise are both much higher than those of Seattle or Portland. This means that the unit for Seattle or Portland can be smaller than is necessary in Spokane or Boise for the same type of house. On the other hand it must operate more of the time especially in early spring or late autumn, and may even be needed occasionally during the summer months because the weather on the Coast is tempered throughout the year by the nearness of the ocean.

Possibilities of Determining Soil Constants: With continually recorded temperatures at various depths below the soil surface as were taken for this study, the transient behavior of the flow of heat can be determined. Since the rate at which the heat travels through the soil is determined by such soil constants as

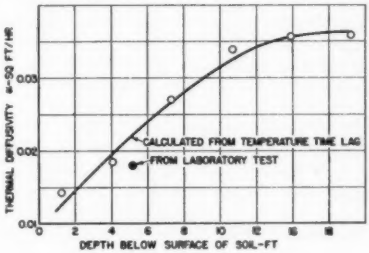


FIG. 10. PLOT OF SOIL THERMAL DIFFUSIVITY AGAINST DEPTH. CALCULATED BY MEANS OF THE TRANSIENT TIME LAG OF TEMPERATURE CHANGE IN THE SOIL

thermal conductivity and diffusivity, such constants can be thus calculated by using equations based upon the solution of Fourier's differential equation[1] for temperature variations with the flow of heat.

Calculations of this nature were made using the formula,

$$\alpha = x^2 P / 4 \pi t^2 \quad \ldots \ldots \ldots \ldots \ldots \quad (2)$$

where

t is the time required for a given temperature to travel x distance either upward or downward.

The subsurface temperatures recorded at Station A were used to determine such time delays. Fig. 10 is a plot of the calculated thermal diffusivity at various depths below the surface. Laboratory tests[2] on a sample taken about 5 ft below the surface at Station A gave the value indicated on the curve as a laboratory determination.

The diffusivities determined by the use of Equation 2 appear to be a little high especially at the greater depths. However, the greater moisture content and possible increased density at such depths would account for some increase.

The temperature records were not taken with this application in view. The use of more accurate methods of recording temperatures for this purpose might have resulted in even better results.

CONCLUSIONS

This study proposes approximate methods of using Weather Bureau records, which are quite generally available, to determine some of the desirable informa-

tion for the design of heat pumps and ground grids. To determine more accurately the ground heat available, the temperatures near the surface of the soil must be obtained throughout the year, and even these would give information for that year only. Needless to say, the time and cost would in general be prohibitive.

If similar comparisons could be made between the approximate methods and the more accurate ones wherever actual ground surface temperatures are avail-

TABLE 6—PERCENT OF HEAT PUMP OPERATING TIME BOISE, IDAHO[3]

MONTH	DEGREE DAYS	PERCENT OF TIME OPERATING[a]	HOURS OF OPERATION	PERCENT OF TIME OPERATING					
				House I	House J	House K	House L	House M	House N
Jan	1097	62	461	64	66	55	85	64	75
Feb	854	53	356	54	42	35	55	75	44
Mar	694	39	290	36	24	23	24	43	37
Apr	438	25	180	17	19	11	8	17	18
May	208	11	82	5	7	4.5	2.3	1	5
June									
July									
Aug									
Sept	87	5	36	0.4	3	4	2	—	1.3
Oct	406	23	170	21	48	11	34	27	30
Nov	728	42	302	26	36	24	42	36	36
Dec	1010	57	424	41	48	42	57	55	57
TOTALS	5522		2301						

DETAILED INFORMATION ON HOUSES

HOUSE	GROSS HEATED CU FT	UNIT	HEAT SOURCE
I	9400	3 hp	Well
J	14300	5 hp	Well
K	10300	5 hp	Well
L	17500	5 hp	Well
M	18900	5 hp	Well
N	24100	7½ hp	Well

[a] Based on average of the coldest January day for seven years, taken every third year. Degree days of average coldest day= 57

able, the accumulated results would probably suggest a very acceptable procedure until enough experience with heat pump and ground-grid operation has been acquired to suggest better methods.

In general climatology data are very valuable in such heating design work.

ACKNOWLEDGMENTS

This paper is a report on one phase of a rather extensive research program supported by grants from the Bonneville Power Administration and the University of Washington Engineering Experiment Station. The author also wishes to mention the valuable assistance of Dr. Phil E. Church, head of the Meteorology and Climatology Department at the University of Washington.

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2. Thermal Conductivity of Soils for Design of Heat Pump Installations, by G. S. Smith and Thomas Yamauchi (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 355).
3. Field Tests and Operating Data on Twenty-Two Heat Pumps in Idaho, by C. E. Boggs (presented to *Northwest Electric Light and Power Association*).
4. Ground Temperatures as Affected by Weather Conditions, by A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 363).

DISCUSSION

L. R. INGERSOLL, Madison, Wis. (WRITTEN): I want to compliment Mr. Smith on his heat pump investigations, partly because of the use he has made of heat conduction theory in connection with his experimental work. It is true that theory is sometimes far from satisfactory here, due to unknown or complicating factors, *e.g.*, moisture migration in the soil, but in any case it can help guide investigation. Without it, experimentation may degenerate into merely "shooting in the dark."

It might be pointed out that a series of underground temperature measurements, such as were made here, afford one of the best possible means of measuring the thermal diffusivity of the soil, and, indirectly, its thermal conductivity. This presupposes that great care is taken—as I assume was done here—in arranging the thermocouples so that conduction along the wires cannot vitiate their readings.

I am a bit interested in the calculation of the total seasonal heat taken in by the ground because of some computations I have myself made, although they have a somewhat different slant. I would warn against overestimating the importance of such calculations, however, because they apply with any exactness only to the hypothetical case of a continuous grid over, or rather under, a large ground surface area. In the usual practical case of more or less isolated pipes a large share of the heat is drawn laterally. Mr. Smith takes account of this fact, but does not stress it as much as I would be inclined to do.

E. B. PENROD*, Lexington, Ky. (WRITTEN): The approximate method, outlined by Professor Smith in his paper, of using Weather Bureau records to determine information needed to design earth heat pump systems is very timely. The proposed method should be utilized by heat pump engineers in designing pilot heat pumps in localities where the necessary data are available. If this is done, operating data should be taken over a number of years to check the method. At the present time there may be sufficient air temperature data in the various weather stations, but little or no surface temperature data have been taken and recorded. Where surface temperature data are not available, the proposed method is not applicable.

The paper will be exceedingly valuable if it stimulates meterologists and climatologists to extend their investigations to the measurement of soil temperatures at various depths and the vertical component of solar radiation. Wherever soil temperatures are measured, the soil should be analyzed to determine the Casagrande classification, moisture content, liquid limit, plastic limit, and the plasticity index.

The author states that air temperatures were recorded from near the earth up to a height of 150 ft at twelve minute intervals, but failed to mention whether the average air temperatures shown in Fig. 4 included readings at more than one elevation.

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* Head, Department of Mechanical Engineering, University of Kentucky.

In discussing the optimum depth for a ground coil, the author did not consider the performance when the heat pump operated on the cooling cycle. In many localities, the earth heat pump will operate as long on the cooling cycle as on the heating cycle. It should be recognized that considerable heat can be stored in the soil during the cooling season provided the earth heat exchanger is placed at the proper depth.

It is interesting to note, in the example cited, that the heat absorbed by soil due to rainfall was approximately 10 percent of that from solar radiation. From Fig. 9, however, for a base temperature of 48 F, it would appear that the heat from the sun should be about 6,400 Btu per sq ft per season instead of 7,450. If so, a value of 6,930 would be obtained. This in turn, would give a column of about 30 instead of 35 ft.

In conclusion, I would like to point out that the proposed approximate method can be understood and applied by a competent design engineer. A casual study of the paper shows that a tremendous amount of work was done in taking and recording the data presented. Heat pump engineers are very fortunate in having the results of this investigation made available.

T. H. URDAHL, Washington, D. C. (WRITTEN): This paper is timely in that it adds further needed emphasis upon the value of a United States Weather Atlas. Such a volume is about to be prepared by the United States Weather Bureau, with the assistance of interested groups if funds are made available by the Congress of the United States.

Data, upon which to base such an Atlas including those which would show the feasibility of heat pump operation in a given locality, are available in present records, but have never been presented in a form readily usable by designers. The processing of data, and its translation into forms generally approximating those indicated by the author, would do much to take the heat pump application into wider practical use.

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been foremost among engineering groups in advocating the compilation and publication of a Climatic Atlas and in showing the immense economic benefits to be realized through a more complete knowledge of climatic conditions, quantitatively expressed for the country as a whole and for localities individually.

The work of the Society, under its TAC on Weather Design Conditions, has stimulated the interest of other groups, and the Weather Atlas has been endorsed as a primary need of the construction industry by *The Building Research Advisory Board* of the *National Research Council, The American Institute of Architects,* the *Air Conditioning and Refrigerating Machinery Association,* and many other trade and professional groups.

C. F. KAYAN, New York, N. Y.: I have a brief question concerning the practical matter of installing the temperature resistance thermometers at different depths. How were they brought down to the different depths, and how were they located at the proper points and protected from extraneous temperature effects.

H. M. HENDRICKSON, Seattle, Wash.: Since Professor Smith has pointed out that the amount of heat available for the heat pump from the ground depends upon the amount of heat picked up in the off season, I am wondering about the character of the surface of the ground, and how that would affect the heat pickup in the off season? I would like to ask Professor Smith if the surface of the ground had vegetation, whether it was a lawn or whether it was ordinary soil, and how the surface would affect the amount of heat that will be picked up by the heat pump during the off season.

J. R. JAMIESON, Urbana, Ill.: I have one question regarding Table 2, Calculated Rainfall Heat Absorbed by Soil. I wonder if that calculation is not based upon the assumption that all the rainfall passes down through the soil, and is taken away

by the movement of the ground water? If that is the case, maybe we are showing a little more than actually would occur because of two factors that would affect it. Seemingly the rainfall hits the soil, and there might be considerable evaporation back to the atmosphere which means the heat would not then be available to the coils in the earth. The soil would be affected by the drybulb temperature of the air, the wind velocity and the relative humidity following any rainfall.

The second factor would be the runoff. In other words how much runs off rather than goes down and is carried away by the ground water? If those two factors are of any magnitude, it might mean that the values given in Table 2 might be somewhat high.

R. C. CHEWNING, Portland, Ore.: Professor Smith, in determining the amount of heat that might be put into the ground, due to rainfall, sun and so forth, is assuming an average ground temperature of 48 F. When a heat pump with a ground coil is in operation, the temperature of the ground in the vicinity of the coil is approximately 25 to 30 F. The lower temperatures in the soil have two important effects: (1) the amount of heat available to the soil from rainfall would be increased several hundred percent, (2) the time lag in the transmission of the heat from the surface down to the coil would be greatly reduced.

Further, since the temperature at which we get the bulk of our rain in this area, is about 45 F, the statement that it is necessary to pick up the heat during the summer in order to operate a heat pump is questionable. I think actual experience does not confirm that conclusion. There is no doubt that it is necessary to bring the ground temperature back up to a higher level during the summer for a considerable distance around the coil, but that is usually accomplished long before the summer is over, probably by the middle of July. From then on the additional heat that is being given by the sun, rain, etc. has relatively small effect on the storage capacity of the soil.

AUTHOR'S CLOSURE: Dr. Ingersoll has called attention to a most important point in conducting research of this type. Without the guidance of mathematical theory, the important conclusions to be checked by experimental work would quite often be completely lost because of the multitude of factors influencing such experimental work.

In further answering Dr. Ingersoll as well as many of the remaining contributors in this discussion, the method of predicting the amount of heat available from a given area was not proposed as an accurate one, but due to the almost complete absence of any other method for making even an intelligent guess, this method bases an estimate upon a logical mathematical basis, with the use of information which can be obtained for any locality. Every effort was made to arrive at results which would be conservative and probably less than actual experience would prove.

Dr. Ingersoll suggests the possible effects of lateral heat flow, Professor Hendrickson mentions the effect of the surface vegetation, Mr. Jamieson and Mr. Chewning mention the effects of rainfall. To satisfy the condition known to prevail at a given site intelligent corrections should be applied for the given case. If there is no other ground grid within a reasonable distance, the perimeter of the area to be assumed as furnishing the heat could be five to ten ft distant from the outside tube in the grid. If the vegetation is dense less heat would penetrate into the soil. If warm winter rains are the rule as in the Pacific Northwest, a considerable additional heat replacement during the heating season could be assumed, and a rough calculation from average weather records could be made. Probably a more accurate estimate of the actual heat available must finally depend upon actual heat pump experience in each area.

The lateral heat flow and especially the heat obtained from the earth's center are of rather minor importance. However, where these is underground water movement, entirely different results might be expected.

In reply to Professor Penrod, the average air temperatures used were purposely those taken by the U. S. Weather Bureau at a station having temperatures very simi-

lar to these at the University. The air temperatures at the University Station were only used for comparison.

Professor Penrod also suggests the consideration of cooling cycles when determining the depth of ground grids. In the Pacific Northwest the cooling cycle is very lightly considered but where cooling is an important factor, the coil depth must be altered. For heat storage a greater depth is highly desirable, and such a depth might give better results for both heating and cooling, when the direct heat absorption during the summer would become a lesser factor.

Mr. Urdahl's mention of a United States Weather Atlas is most interesting. Such a publication would have been most valuable in a study of this type. If surface and subsurface temperatures at various parts of the country had likewise been available, comparisons of the calculations for the surface and maximum air methods could have been carried out for several locations. Similar calculations made for at least one other very different location would be most desirable.

In answer to Professor Kayan, the resistance thermometers were fastened to a light wood support, placed in the well and the well filled in.

I have already suggested the effect of surface foliage mentioned by Professor Hendrickson. The surface temperatures obtained for this study were purposely taken on a site with a normal clipped lawn and with a few trees which would cast shadows during portions of the day. This was intended to correspond to the average lot under which a ground grid might be placed. Studies made with bare ground as compared to considerable vegetation do indicate a somewhat lesser absorption with a moderate amount of foliage but unless the covering is quite dense the results are not radically different.

Both Mr. Jamieson and Mr. Chewning stress rainfall. The proposed calculations for rainfall are far from satisfactory, and run off as well as evaporation could easily detract from that calculated. However, I must agree with Mr. Chewning that rainfall, especially during the heating season, and especially in the Pacific Northwest probably makes a very marked contribution to the available heat for a ground grid. In my calculation I assumed no heat from rainfall when the average air temperature was equal to or below the assumed average soil temperature and this eliminated almost all the heating season. In general rain during the heating season occurs at times of abnormally high temperatures compared to the average for that season, and since water has a very high specific heat the heat carried into the soil during winter rains is probably of considerable importance. To calculate the amount would usually require more information than is often available in the usual weather reports. However the necessary information can usually be supplied by the Weather Bureau upon request.

I cannot quite agree with Mr. Chewning that the average temperature of the whole plot drawn on by a grid would be as low as 25 to 30 F unless severe freezing of the soil was allowed. Such severe freezing would cause severe heaving of the soil and perhaps damage to surface foliage.

1436

No. 1436

HEAT REQUIREMENTS OF BUILDINGS

By C. W. Signor,* Detroit, Mich.

THE PURPOSE of this paper is to make available a tabulation of the heating and process steam requirements of 544 buildings located in the business center of Detroit, Mich. These buildings are grouped into 24 classifications according to their usage and size and are supplied with steam from the underground district steam distribution system. The maximum steam pressure delivered to the user is 28 to 32 psig with the exception of a 100 psig main serving the high pressure requirements of two hospitals. The steam enters the buildings through a service pipe and then the pressure is reduced as required for heating and process.

Steam Metering

All condensate from the heating and process equipment in a building is returned to a central point, where possible, and it then goes to a condensate meter. The amount of steam used in sterilizers, cookers, clothes presses, etc., and which cannot be metered is relatively small and is not included in the figures in this paper. In such buildings as hospitals and hotels, where considerable open end steam is used, this steam is metered by a flow meter.

Buildings and Equipment

The buildings are located in or near the downtown business district of Detroit. These buildings vary in type in this area from some of the older to some of the most modern.

In nearly all of the heating systems, steam is used in the radiators, but in a few cases steam is used by means of a converter to supply hot water for space heating. The steam heating systems vary from the old one-pipe gravity heating system to a modern two-pipe high vacuum, zoned heating system.

In some of the buildings, heating is still controlled by manual operation of gate valves. In the remainder many types of building temperature controls are

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* Supervisor of Customers' Steam Service, The Detroit Edison Co. Member of A.S.H.V.E.

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used, including unit thermostats, outdoor actuated cycling controls, outdoor actuated program controls, and the medium and high vacuum controls. Buildings have one or more controls depending upon the number of zones.

STEAM CONSUMPTION RECORDS

There were 1786 customers being served by the distribution system during the heating season of 1948-49. For each building with more than 750 sq ft of radiator surface, a *Steam Consumption Record* card such as shown in Table 1 was prepared. Each month the amount of steam used by each building and the cost is entered on the card.

The *Estimated Monthly Consumption for Process, Lb,* as shown on the card, is metered in a few buildings, but for most of the buildings it represents the

TABLE 1—STEAM CONSUMPTION RECORD

RADIATION 20,120 — SPACE CU FT 1,315,000

HEATING SEASON 1948-49 — HEATING SEASON 1949-50

NAME Office Building — METER NO

ESTIMATED MONTHLY CONS FOR PROCESS, LB 40,000 — BILLING DEMAND

PERIOD ENDING	TOTAL CONSUMPTION LB	CONS FOR HEATING	AVG MEAN TEMP	D D 65 F BASE	LB/DEG DAY 65 F BASE	LB/DEG DAY F CU FT	NET BILL PLUS SALES TAX
SEP 48	51300	11300	66	74			77.91
OCT 48	458000	418000	51	444	941	.318	684.39
NOV 48	547600	467600	42	579	820	.623	657.73
DEC 48	908000	863000	32	981	879	.669	1129.96
JAN 49	948000	858000	29	965	893	.677	1134.83
FEB 49	774000	734000	28	1011	814	.619	1006.80
MAR 49	704000	564000	36	880	646	.466	798.43
APR 49	440000	400000	51	638	510	.343	568.62
MAY 49	164000	124000	57	214	594	.227	254.38
JUN 49	84000						99.69
JUL 49	38000						42.53
AUG 49	34000						37.87
TOTAL	5211900	4764300		6377	818	.625	6666.73
1947-48	5899000	4804600		6579	914	.698	6444.50

PERIOD ENDING	TOTAL CONSUMPTION LB	CONS FOR HEATING	AVG MEAN TEMP	D D 65 F BASE	LB/DEG DAY 65 F BASE	LB/DEG DAY F CU FT	NET BILL PLUS SALES TAX
SEP 49	113000	89000	66	180			166.71
OCT 49	212000	182000	53	432	674	.513	337.88
NOV 49	796000	716000	41	784	913	.694	820.96
DEC 49	916000	886000	28	968	915	.674	1132.78
JAN 50	899000	849000	28	984	862	.643	1107.63
FEB 50	826000	786000	36	828	949	.718	1016.45
MAR 50	590000	550000	34	846	650	.494	718.51
APR 50	411000	371000	46	729	509	.387	506.49
MAY 50	202000	172000	57	396	434	.330	257.21
JUN 50	50000						59.09
JUL 50	40000						46.83
AUG 50	37000						43.32
TOTAL	5840000	5439000		6269	868	.658	7393.46

average monthly steam consumption during the summer months when building heating is not required. *Process* includes water heating, cooking, etc.

Consumption for Heating is the difference between the total consumption and the process load.

Average Mean Temperature and the *Degree Days, 65 F base* are from the data of the United States Weather Bureau located at the Detroit City Airport.

Lb per Degree Day and the *lb per Degree per M cubic feet of Space* are computed and entered each month when heating is required.

The cubic feet of space for each building, as is shown at the top of the card (Table 1) is not the architectural cubage but is the heated space from inside measurements of the building.

PURPOSE OF STEAM CONSUMPTION RECORD

The purpose of keeping steam consumption records for these buildings is to obtain an indication whether the use of steam is above or below the normal requirements of the building and whether the use is above or below normal for the type of building or business.

When the steam use in a building is below normal, it is usually due to a reduction in: space heated, hours of heating, or inside temperature, or it may be due to loss of condensate before metering. When the steam consumption is

above normal it is usually due to increased space heated, increased hours of heating, careless operation of equipment, or faulty equipment.

The building owner or engineer is notified of abnormal use of steam and advised how waste can be eliminated. This requires a continuous educational

TABLE 2—STEAM CONDENSED PER SQUARE FOOT OF RADIATOR HEATING SURFACE[a]

YEAR	AVG TEMP HEATING SEASON, F	LB/SQFT RADIATOR SURFACE	LB/SQ FT RADIATOR SURFACE/DEG BELOW 65 F
1921	42.5	473	21.1
1922	40.8	470	19.4
1923	37.8	484	17.8
1924	38.6	484	18.4
1925	38.2	467	17.5
1926	36.1	493	17.1
1927	39.2	418	16.3
1928	39.9	436	17.4
1929	40.4	436	17.7
1930	39.4	366	14.3
1931	40.7	303	12.5
1932	43.2	308	14.2
1933	40.4	295	12.0
1934	37.2	315	11.4
1935	38.4	316	11.9
1936	37.4	372	13.5
1937	38.8	329	12.6
1938	40.0	290	12.3
1939	40.2	307	12.3
1940	37.2	338	12.0
1941	39.9	294	11.7
1942	41.7	318	13.7
1943	37.8	358	13.2
1944	39.1	332	12.8
1945	39.5	353	13.8
1946	40.3	323	13.1
1947	39.4	362	14.2
1948	39.0	360	13.8
1949	41.6	328	14.0

[a] Based on the average of total connected radiator heating surface at the beginning and ending of each year.

program in which users are informed about the best and most economical ways of using steam service.

The results of work with users is illustrated in Table 2 which is based on the total pounds of steam consumed by all users during a year, the average connected radiator surface and the average temperature for each heating season. The last column, Pound Per Square Foot of Radiator Surface Per Degree below

TABLE 3—HEATING AND PROCESS STEAM REQUIREMENTS OF BUILDINGS

CLASSIFICATION	NUMBER OF BUILDINGS	TOTAL CONSUMPTION		HEATING CONSUMPTION		DEGREE DAYS	HEATING ONLY LB STEAM/DD /M CU FT OF HEATED SPACE	PROCESS ONLY LB STEAM /CU FT OF HEATED SPACE/YR
		Lb Steam/Sq Ft of Rad Sur/Yr	Lb Steam/Cu Ft of Heated Space/Yr	Lb Steam/Sq Ft of Rad Sur/Yr	Lb Steam/Cu Ft of Heated Space/Yr			
Apartments	25	584.6	9.50	412.3	6.70	5797	1.15	2.79
Apartment Hotels	4	394.2	9.11	261.0	6.03	5797	1.04	3.07
Banks	16	208.4	4.43	178.5	3.80	5797	0.656	0.635
Bowling Alleys and Sports	7	188.6	3.85	176.2	3.59	5797	0.620	0.252
Churches	17	190.1	3.35	177.5	3.13	5797	0.541	0.222
Clubs	16	343.1	7.89	214.0	4.92	5797	0.850	2.97
County, Federal and State Building	24	239.7	4.33	214.2	3.86	5797	0.667	0.464
Garages[a]	10	307.1	3.12	286.2	2.90	4250	0.684	0.211
Hospitals	6	736.9	12.44	392.3	6.62	5797	1.142	6.20
Hotels—Small[b]	34	586.6	9.66	406.8	6.64	5797	1.15	2.93
Hotels—Large[c]	17	668.7	10.30	384.5	5.92	5797	1.02	4.37
Manufacturing Concerns	53	356.0	5.61	317.0	4.99	5797	0.859	0.613
Miscellaneous	2	156.9	3.65	129.5	3.01	5797	0.520	0.633
Newspaper & Printing Est.	22	427.0	7.71	362.6	6.52	5797	1.12	1.15
Office Bldg—Small[d]	78	321.3	5.41	284.4	4.79	5797	0.825	0.625
Office Bldg—Medium[e]	20	307.4	5.06	250.4	4.12	5797	0.714	0.940
Office Bldg—Large[f]	20	259.1	4.43	211.8	3.61	5797	0.623	0.808
Restaurants	19	291.2	14.57	116.4	5.81	5797	1.00	8.73
Sales and Service	15	302.9	4.90	268.1	4.34	5797	0.750	0.561
Schools	8	123.8	4.76	110.5	4.25	5797	0.734	0.509
Theatres	24	139.9	2.91	132.6	2.76	5797	0.477	0.152
Stores—Retail	87	220.3	3.94	147.8	2.64	5797	0.456	1.29
Stores—Wholesale	15	300.5	4.35	275.7	3.99	5797	0.690	0.358
Warehouses	10	324.2	3.52	308.9	3.35	5797	0.579	0.174

a A degree basis of 65 F was used for all buildings except garages which are based on 55 F basis
b Hotels under 1,000,000 cu ft of heated space
c Hotels over 1,000,000 cu ft of heated space
d Office buildings under 1,000,000 cu ft of heated space
e Office buildings between 1,000,000 and 2,000,000 cu ft of heated space
f Office buildings over 2,000,000 cu ft of heated space

65 F shows considerable decrease between 1921 and 1931. In the years from 1932 to the present this figure has fluctuated somewhat due to various factors such as economic conditions and growth of system.

HEATING AND PROCESS STEAM REQUIREMENTS OF VARIOUS BUILDINGS

The period covered by this study was the 12 months ending August 30, 1949. The steam consumption records for buildings having more than 2000 sq ft of radiator surface were classified into 24 different groups as shown in the left hand column of Table 3. The steam consumption for the buildings in each group has been calculated on the basis of radiator surface and heated space. Steam for heating has been given in pounds per degree-day per thousand cubic feet of heated space. Steam for process is given in pounds per cubic foot of heated space per year.

DISCUSSION

D. W. LOUCKS, Pittsburgh, Pa. (WRITTEN): This paper contains much information of interest and value to those concerned with the operation and maintenance of buildings.

In the first place, Table 1 shows a method of keeping a record of steam consumption. Many if not all, district heating companies keep a steam consumption record of all or most of their steam users in one form or another in order to assist the users to attain economical operation. It affords a means of watching the performance of the heating system and the care with which it is being operated. Everyone interested in economical operation can profit by establishing some system of recording and observing the rate of steam use.

This paper also shows the value of efforts to reduce steam consumption as indicated in Table 2. By averaging the rates of steam use for each five year period the lbs per sq ft of radiator surface per degree below 65 F is as follows:

1921-25—18.8	1936-40—12.5
1926-30—16.5	1941-45—13.04
1931-35—12.4	1946-49—13.8

It will be noted that the rate of steam use was highest during the period 1921-25, being 18.8 lbs per sq ft, and lowest during the five year period 1931-35 and then started upward again from this period. This is an interesting trend because it indicates that about ten years of concentrated effort on the part of the district heating company was required to reach the maximum effect of such efforts over a large group of steam users. The gradual increase after that period up through 1949 is undoubtedly caused by improved business conditions. Improved business conditions may cause an increased rate of consumption in two ways—first, the occupancy and the activity in any given building increases, and secondly, as business improves there is a lesser tendency to scrutinize such costs as heating as carefully as might be the case under poor business conditions.

The information given in Table 3 indicates certain unit measures of steam usage for various classes of properties. It should be pointed out that these measures are more relative than absolute. In other words, these rates of steam use should be used more for comparison purposes than as a means of estimating steam consumptions for various buildings.

The rate of steam use in pounds per cubic foot per year is not an accurate method of estimating steam usage because of the influence which the shape of the building has on the ratio of volume to exposed wall area and also for that reason it is not a highly accurate method of comparing the steam consumptions of one building with

another. For example, one building having dimensions 25 ft × 100 ft has the same volume as another building having dimensions of 50 ft × 50 ft, provided they have the same height. In one case the perimeter of the building is 250 ft and in the other case the perimeter is 200 ft. It would, therefore, be expected that the rectangular building would have a greater use per unit of volume than would the square building.

While the unit steam consumption per square foot of installed radiator surface may be more accurate, it also is subject to error because the amount of actual radiation installed may not conform to the actual or calculated heat loss requirements. This condition is frequently encountered in buildings of the older type.

These methods of measuring the rate of steam usage are, however, most valuable in watching the year to year performance of a given building and it is also valuable if used advisedly as a guide to the relative efficiency with which buildings of the same class are using steam.

AUTHOR'S CLOSURE: I appreciated the detailed and enlightening discussion of my paper by D. W. Loucks. I do not feel that it is necessary to review his comments except to say that I am in complete agreement with them.

However, I would like to emphasize that the figures in the paper represent average use of steam for heating and process for each class of building and the intended use of these figures is to indicate when the use of steam for a building is above or below normal.

Even though the steam use for any building may vary considerably from these average figures, it can still be normal for the conditions existing in the building.

The steam used for heating will be influenced by the hours of heating, the lighting load, the condition and maintenance of the building and heating system, the type and amount of occupancy in the building, the temperature controls and, as Mr. Loucks pointed out, the relation of the outside exposure to the volume.

The steam used for process will be influenced by the type and amount of occupancy, the amount of business, the controls and maintenance of the process equipment.

With complete understanding of the factors which influence the amount of steam used, the figure in Table 3 can be used for rough estimates of steam use.

In closing, I would like to thank Mr. Loucks for his discussion.

1437

No. 1437

ENERGY—CHOOSE IT WISELY TODAY FOR SAFETY TOMORROW

By G. W. GLEESON*, CORVALLIS, ORE.

IN THE concept of the broad divisions of the entity of engineering, we include men, money, methods, materials, manufacture, and energy. It is significant that we separate and fragment each broad classification to our probable detriment, but, basically, the study of energy as a whole subject is fundamental to any and all engineering patterns. When considered from the viewpoint of the attributes of our western civilization all else depends upon our ability to utilize our energy resources. Without adequate energy resources, manufacture cannot take place, consideration of methods becomes insignificant; material cannot be processed for use; money will not circulate, and men will not prosper according to our present day standards.

It is common practice to study energy in a fragmented part, even to the second and third degree of fragmentation. For example, we circumscribe a field of thermodynamics for mechanical engineers, calling it, perhaps, fuel and combustion, or heat power, but we do not include within the boundaries the total subject matter pertinent to even the first breakdown. We fragment further to include only that portion of thermodynamics or other subject matter which a mechanical engineer *commonly* uses, and then, because we have lost the fundamental continuity of a whole pattern, we resort to empiricism and tabular or graphical means of calculation which are restricted to single working substances or fixed conditions. The more we fragment, the more we limit the range of the subject and straight-jacket our thinking.

If this paper does nothing more than raise the singular subject of energy above the detail of its fragmented parts, it will have served a purpose. However, it is intended to evaluate energy as a resource, irreplaceable in large part, and deal with the resource upon the same basis as we would our resources of material. The *conservation of energy* is more than a law of physics or principle

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of thermodynamics. Our dependence upon our energy resources are perhaps even more significant than our dependence upon material, but our consideration of such resources appears to be inversely proportional to our use.

Accordingly, that which follows attempts to consider all energy resources, as it is highly probable that the pattern of energy usage in the not too distant future will involve an appreciable amount of substitution of sources, either directly or through new process development. It is pertinent to mention that considerations of energy alone may indicate possible shifts of population centers and most certainly shifts in industrial concentration. It is certain that more complete integration of energy uses compatible with domestic or industrial requirements will be necessary and will be dictated by economic considerations if not by conservation practice.

The writer is particularly indebted to many previous authors and all sources of information as enumerated in the appended bibliography.

General Considerations

As would naturally be the case as dictated by economy, man has made greatest use of those energy resources which were most accessible or *packaged* in the most desirable physical state. Accordingly, our so-called fossil fuels, including coal, petroleum, and natural gas have been most widely exploited. Energy sources may be classified for considerations of conservation into two broad categories: (1) Continuous sources of energy which can be used to maximum possible extent without depletion; and (2) discontinuous sources of energy which have definite, although somewhat unknown, limitations in amount. The fuels of greatest current use fall within the discontinuous classification; hence, the prospect of ultimate depletion of reserves of the discontinuous type become a reality under conditions of current rate of use.

Enumerating according to the classification:

(1) Continuous Sources of Energy: a. Water Power; *b.* Energy from Vegetation (directly and indirectly); *c.* Solar Energy; *d.* Wind Power; *e.* Earth Heat; *f.* Tidal Power; *g.* Tropical Waters; *h.* Heat Pump.

(2) Discontinuous Sources of Energy: a. Coal; *b.* Petroleum; *c.* Oil Shales; *d.* Natural Gas; *e.* Atomic Energy.

In the following, each of the energy sources will be discussed briefly, particularly from the standpoint of limitations. Such discussion must, obviously, be confined by present day concepts and be without the benefit of such future developments as man is certain to bring forth. It is significant that most authorities agree that the major advances of the future will be in the field of continuous energy supplies. Yet, at the present time, attention is almost exclusively directed toward our discontinuous sources, and, because we are more familiar with them, such sources will be given priority in order of discussion.

Before a consideration of individual fuels, it is pertinent to provide a frame of reference in terms of the estimated overall world resources of energy as set forth in Table 1 in equivalent terms of millions of tons of bituminous coal.

It appears to be a paradox, but, in terms of the present direction of technical evolution, Asia, particularly China, and North America have the greatest potentialities for progress. It is necessary that America maintain a rate of capital

growth sufficient to offset the energy and population potentials of Asia if we are to maintain our dominant position in world affairs. This statement stems from the commonly recognized fact that national power depends upon three major components, population, energy resources, and the application of capital or its equivalent.

Furthermore, as a general consideration, we cannot escape our obligation or responsibility to future generations. Repudiation of such responsibility robs life of its significance, although we seldom take time to consider the full meaning of our transient existence. Certainly, we of the present generation are, in part,

TABLE 1—WORLD RESOURCES OF ENERGY

AS MILLIONS OF TONS OF BITUMINOUS COAL	
North America	3,087,143.6
South America	166,881.0
Europe[a]	789,548.8
U. S. S. R.	1,313,685.3
Asia[a]	2,347,544.1
Africa	765,005.3
Oceania	190,477.7
World	8,660,285.8
SELECTED COUNTRIES	
United States	2,513,693.4
China	2,158,840.0
Germany	308,170.5
South Africa	210,370.0
British Isles	177,047.5
Australia	141,158.0
India	99,754.6
Poland	97,380.5
Japan	41,658.2
France	33,472.0

[a] Exclusive of U.S.S.R.

guardians of the future; we simply have lease upon the *stuff of life,* but we appear bent upon consumption of the leaseholding. Most certainly. future generations have some equity in that which we so unthinkingly consume today, hence, conservation in its broad aspects has far reaching ramifications. Among the many things we should harbor and conserve, perhaps none are more important than our readily available, discontinuous sources of energy.

COAL

Confusion exists as regards coal reserves. The estimation of any underground resource can never be exact. Many estimations of the past have been construed as reference to proved reserves, and many estimates have not been classified as regard to cost of recovery or, for that matter, actual feasibility of recovery. We may be sure, however, that our coal reserves represent our most extensive dis-

continuous energy resource. Two extensive federal surveys have been made in 1907 and 1936, but due to marked variations which have appeared in recent years, another survey is underway that will be more concerned with the practical problem of recoverability. We are currently uncertain of our potential, therefore, and must wait several years for the more comprehensive and complete picture. Table 2 gives selected results from the two previous federal surveys.

The appreciable decline in magnitude of estimates between the two surveys may be noted in Table 2.

TABLE 2—FEDERAL COAL RESOURCES SURVEY[1]

IN BILLIONS OF TONS OF COAL

STATE	1907	1936
North Dakota	500	600
Wyoming	424	620
Montana	303	381
Illinois	240	198
West Virginia	231	147
Utah	196	93
New Mexico	164	21
Pennsylvania	134	118
Missouri	40	84
Michigan	12	2

TABLE 3—BILLIONS OF TONS OF COAL[1]

STATE	1936 FEDERAL SURVEY	STATE SURVEY CORRECTED
Pennsylvania	118	44
Ohio	92	10
West Virginia	147	116
Utah	93	46

Various states have made their own surveys with suitable corrections for unrecoverable material, production, and losses. Table 3 presents some comparative results.

We have come to believe that our coal reserves are inexhaustible, but even a superficial study indicates wide divergence of estimated reserves from the availability viewpoint. Estimates vary from 200 to 1500 years supply as based upon the current total annual energy requirement of the U. S. Probably the actual value lies somewhere between the two extremes. The assumption of even the minimum period of time may result in complacency; however, a few disturbing features are evident.

We can be certain that coal will become more costly to produce and more costly to use. It will come from thinner and less accessible seams in more remote areas. It will have higher ash, more sulphur, and lower heat value.

[1] Exponent numerals refer to References.

The weighted average thickness of coal seams has been estimated at 5.3 ft, and, at such thickness, one year's production of coal would exhaust about 120 sq mi. Thicker seams have already been exhausted in the older territories and mining costs are increasing rapidly.

Coal to transport becomes an ever increasing cost factor as shipping distances increase.

No single coal deposit is known that will provide for large-scale production in an immediate, convenient, and economic manner sufficient for liquid fuel production by synthesis.

Underground gasification may prove feasible without large waste.

About 20 percent of the bituminous coal produced goes to coke. Depletion of coking coal reserves has progressed three times as rapidly as others, and it

TABLE 4—WORLD OIL PRODUCTION IN THOUSANDS OF BARRELS PER DAY AVERAGE

AREA	Nov. 1949	Nov. 1950
Western Hemisphere	1932.1	2206.2
Europe and Africa	75.7	87.5
Middle East	1459.7	1900.9
Other Asia	210.3	261.7
Russia and East Europe[a]	859.0	909.1
	4536.8	5365.4
United States	5209.5	5895.0
Total	9746.3	11260.4

[a] Estimated

is generally agreed that a relatively high percentage of exhaustion of coking coal reserves has already been accomplished. Certainly, we should give special consideration to our mis-use of good coking coals in domestic furnaces or industrial steam plants.

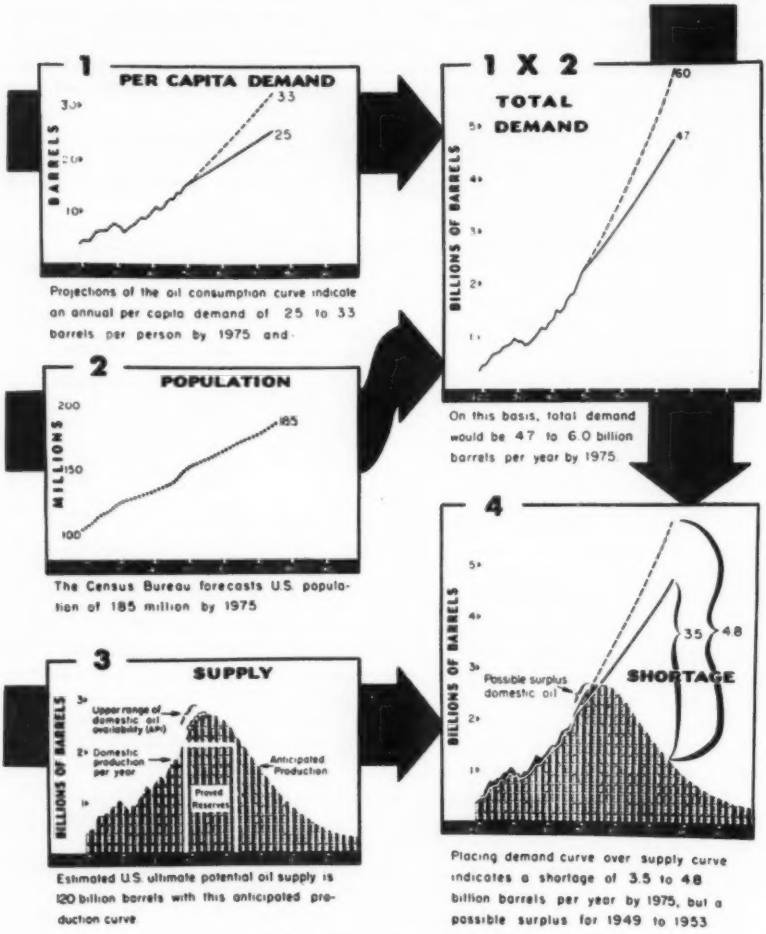
PETROLEUM

Because of the desirable physical state of petroleum products, large scale use has developed to the point where our petroleum reserves are of critical importance. Average production of oil over the world is given in Table 4.

It is evident that the U. S. produces over one-half the oil of the world and consumes the major portion of its production. If we produce as much oil in the future as we have produced in total up to the present, we will be fortunate. A 100-million-barrel field is now an exciting discovery and we shall have to locate the equivalent of about 100 fields of this size to produce as much in the future as we have in the past. Such discovery does not appear probable.

Much could be said about our petroleum situation. Boyd[3], however, has recently presented a concise, graphical summary (see Fig. 1). The projected demand superimposed upon the most reliable estimates of supply indicates a period of petroleum shortage after about 1955, this shortage becoming increas-

ingly acute with time. Some technologists are more optimistic than Mr. Boyd and there is really no way of knowing whether we shall add 10 or 55 billion barrels of oil to our domestic reserves. These two figures represent the extremes.


1
PER CAPITA DEMAND
BARRELS
33
25
Projections of the oil consumption curve indicate an annual per capita demand of 25 to 33 barrels per person by 1975 and
2
POPULATION
MILLIONS
200
150
100
185
The Census Bureau forecasts U.S. population of 185 million by 1975
3
SUPPLY
BILLIONS OF BARRELS
Upper range of domestic oil availability (API)
Domestic production per year
Proved Reserves
Anticipated Production
Eshmated U.S. ultimate potential oil supply is 120 billion barrels with this anticipated production curve
1 X 2
TOTAL DEMAND
BILLIONS OF BARRELS
60
47
On this basis, total demand would be 4.7 to 6.0 billion barrels per year by 1975.
4
BILLIONS OF BARRELS
35
48
Possible surplus domestic oil
SHORTAGE
Placing demand curve over supply curve indicates a shortage of 3.5 to 4.8 billion barrels per year by 1975, but a possible surplus for 1949 to 1953

Courtesy U.S. Bureau of Mines

FIG. 1. OIL PROBLEM IN THE UNITED STATES

Even the most optimistic figure, however, does not prolong the life of our domestic resources greatly, and particularly so if we continue the expansion of petroleum use. Such impending shortage of supply brings several considerations into sharp focus.

In the immediate future we shall probably increase our petroleum imports by an appreciable amount.

To provide our growing motor fuel requirement alone, which was approximately 37 billion gallons in 1949, will, in the not too distant future become a significant problem.

Considering the important uses of petroleum such as for motor fuel, solvents, petro-chemicals, etc., it appears foolhardy to use petroleum as a source of energy for space heating or for transportation motive power when other fuels might be substituted.

Certainly, with regard to our petroleum resources, we have given little attention to the requirements of future generations and have depleted a vital source of energy solely upon the basis of immediate economics and availability, with no thought for the long-time significance of such action.

OIL SHALES

Our oil shale deposits are located mainly in the so-called Green River formation as represented by deposits in Colorado, Utah, and Wyoming. The combined results of the U. S. Geological Survey, the U. S. Bureau of Mines, General Petroleum, and a report upon the Naval Oil-Shale reserves indicates a probable 345 billion barrels of oil, although the deposits have not been surveyed completely. This is about 10 times our total petroleum production up to date, and, as such, represents a vital, undeveloped potential.

Only a relatively meager sampling program has been conducted upon the shale, considering the quantity and extensive area of coverage. Colorado contains approximately 300 of the estimated 345 billion barrels, yet there is only one sample section for every 39 sq mi. In addition, the oil yield from the shale varies greatly from traces, to as high as 60 gal per ton of shale. The principal costs of mining, transport, and retorting are the same, regardless of the yield; hence there must be some cut-off value below which oil recovery would be improbable. This figure has been more or less arbitrarily set as 30 gal per ton. With such cut-off figure, allowing for mining losses, and with due regard for costs of extraction up to a figure as high as four times the current price of competitive crude, it appears that a reasonable reserve of 80 billion barrels may be depended upon.

At current rate of consumption, assuming direct substitution, the oil shale would provide our petroleum needs for approximately 28 years or, assuming minimum projected 1975 consumption figures, for 17 years. It is pertinent to remark that oil from shale presents as yet, grave difficulties in the production of a suitable quality motor fuel. Judicious use of shale oil to extend our petroleum reserves is feasible, however, and may be an accomplished fact when it becomes necessary to import appreciably larger amounts of petroleum. What the production rate could or would be depends upon so many intangible factors that little more can be said, other than that the potential reserves exist and will, no doubt, be exploited.

NATURAL GAS

Natural gas reserves which are mainly located in about 2 percent of the petroleum producing fields are variously estimated at from 175 to 400 trillion cubic

feet. In the immediate past, gas has been discovered faster than it has been produced, due to lack of pipe line facilities. Currently, gas is near the peak of withdrawal for optimum recovery, but there is every evidence that natural gas is being, and will be, consumed at an ever increasing rate, particularly for industrial power. There is no method of accurately estimating the probable period of useful life of known reserves since increases in rate of use is an unknown factor. At current rate of use, however, an outside expectancy of 65 years may

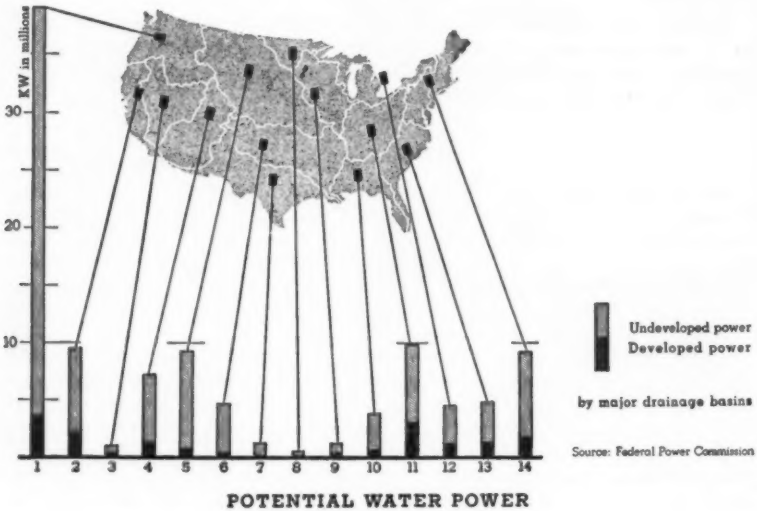


FIG. 2. POTENTIAL WATER POWER IN THE UNITED STATES

1 North Pacific	5 Missouri River	8 Hudson Bay	11 Ohio River
2 South Pacific	6 Lower Mississippi River	9 Upper Mississippi River	12 Great Lakes—St. Lawrence
3 Great Basin	7 Western Gulf	10 Eastern Gulf	13 South Atlantic
4 Colorado River			14 North Atlantic

be the order of magnitude. Most certainly, the actual period will be less than the 65-year figure.

ATOMIC ENERGY

Energy produced by nuclear fission has developed another concept of practically an inexhaustible energy source. This certainly is not true. We have not yet developed controls to the point where atomic energy can be transformed for domestic or industrial purposes, but the accomplishment is not an impossibility. The development of large scale atomic power will come slowly. As with any energy resource, it is dependent upon a *fuel* supply in the form of fissionable material which, like other fuels, is definitely limited in amount and vastly more expensive per unit weight, but undoubtedly competitive per unit of available

energy. Problems of control, energy transfer, protection, poisoning, waste disposal, and *breeder* operation must be solved before practical power generation is a reality. Following the development of the technology of the operation, attention must be given to the *fuel* sources of uranium and thorium which are limited in amount, although not entirely known as yet. Undoubtedly the inventory of atomic *fuel* will be classified information for sometime to come. Even most optimistic estimates, however, lead to the conclusion that nuclear fission as a source of energy will be a highly specialized one, having definite limitations and a probable transitory existence comparable to that of petroleum.

TABLE 5—HYDROELECTRIC POWER IN THE UNITED STATES[4]

DRAINAGE AREA	DEVELOPED POWER JAN. 1, 1960		POTENTIAL POWER	
	INSTALLATIONS	1000 KWHR/YEAR	INSTALLATIONS	1000 KWHR/YEAR
North Atlantic	89	7,407,400	237	27,879,200
South Atlantic	49	3,420,000	143	15,133,500
Eastern Gulf	17	2,781,000	77	15,580,600
Ohio River	46	15,131,500	142	42,219,400
Great Lakes—St. Lawrence	103	7,817,400	228	25,561,400
Hudson Bay	1	19,000	11	110,300
Upper Mississippi	28	1,944,000	131	5,604,300
Missouri	26	3,406,800	172	44,424,300
Lower Mississippi	8	964,000	99	13,806,700
Western Gulf	9	628,000	61	3,607,005
Colorado	17	6,814,650	99	35,838,350
Great Basin	23	934,000	59	3,810,100
South Pacific	70	12,411,000	259	50,716,000
North Pacific	66	23,432,000	645	193,918,700
U. S. Total	552	87,111,350	2,363	478,209,855

It is pertinent to mention that, in one respect, atomic energy is unique. It is the only energy source in the discontinuous category which does not depend upon a fossil fuel, and, accordingly, uses the only fuel which is inorganic.

GENERAL SUMMARIZATION OF DISCONTINUOUS ENERGY SOURCES

There is a mass of detailed information regarding our discontinuous sources of energy which lead to various conclusions. Pertinent factors have been sketched in only a very brief manner. Even from a superficial examination, however, certain important trends may be recognized and stated as follows:

1. The fuels of our discontinuous sources of energy are finite in amount and the *cream* has already been consumed.

2. Our coal reserves are large, but the practicable amounts of recovery depend upon location, nature of bedding, and price. The ultimate, economical, recoverable reserve has not yet been determined.

3. The physical state of petroleum and its role as a portable fuel has caused marked increase in consumption.

4. Oil from shale is a definite possibility, but its competitive position has yet to be determined.

5. Substitution of one fossil fuel for another appears inevitable with the passing of time and synthetic liquid fuels have a definite role to play in the pattern of the future.

6. Extensive importation of petroleum appears to be an immediate consideration.

7. Ultimate life of our fossil fuels is a conjecture and the depletion of one accelerates the depletion of the others. Substitution through synthesis (as oil from coal) involves tremendous energy losses which further accelerate depletion.

8. Within the coming century, the fuel situation as we now know it will not exist.

9. The stage is set for real consideration of our continuous sources of energy.

WATER POWER

The most completely developed of our continuous sources of energy is water power. In the minds of many, particularly the people in the favored northwest area, our water resources are considered all-important. Purely theoretical considerations of all rainfall of the earth (average 36 in. per year) raised to a mean land elevation (2400 ft) and wholly utilized, would produce some 360 trillion horsepower-hours per year. This represents approximately 14 times our present total world energy requirements (aproximately 12 trillion horsepower-hours for U. S. or 24 trillion horsepower-hours for the entire world). Reliable engineering analyses indicate that perhaps not more than 6 trillion horsepower-hours is a practical maximum which might ultimately be developed. Of this practical potential, only about 0.7 trillion horsepower-hours is currently installed. Table 5 presents the installed and potential water power of the U. S. by drainage areas, and Fig. 2 illustrates the same information.

The total U. S. hydropotential of approximately 105 million kilowatts with production of approximately 500 billion kilowatt-hours per year, amounts to only between 5 and 6 percent of our present energy demand and the percentage will decrease as the total national demand increases with the years. It is also evident that the world contribution of hydro to world energy demands will probably not exceed the 6 percent figure. Accordingly, the major virtue of hydro-electric power is the fact that it is a continuous source of energy, essentially *free* from the standpoint of consumption of our natural resources. As such, it should, in the public interest, be exploited to the maximum extent feasible.

PARTIAL RECAPITULATION

Thus far, and in a very brief manner, the *usual* sources of energy have been discussed; coal, petroleum, oil shale, natural gas, and atomic energy under the discontinuous sources and hydroelectric power under the continuous sources. Since these six energy sources provide for current usage, it is well to summarize the national production picture as provided by the U. S. Bureau of Mines figures for 1947. The 1947 figures are essentially those of today with an expansion of usage of petroleum and natural gas at the expense of coal as illustrated by the most recent data presented in Fig. 3. Tables 6, 7, and 8 present the 1947 inventory, with Table 6 giving the individual supply figures by production sources. It must be recognized that part of the production from one source in Table 6 is used to produce energy credited to another source such as coal to manufactured gas or crude oil to refineries. It is necessary to correct Table 6 to avoid *double counting*, and the corrected values appear in Table 7.

TABLE 6—ENERGY PRODUCTION AND IMPORT DATA FOR 1947[5]

U. S. BUREAU OF MINES CIRCULAR 7582
VALUES 10^{15} BTU

SOURCE	PRODUCED	IMPORTED	TOTAL
Bituminous Coal	16,532	8	16,540
Anthracite Coal	1,556	—	1,556
Crude Oil	11,142	585	11,727
Natural Gas	6,074	—	6,074
Natural Gasoline and Allied Products	610	—	610
Coke Oven	2,589	3	2,592
Manufactured Gas	507	—	507
Refineries	11,531	371	11,902
Electric Utility Power	4,364	22	4,386
Total	54,905	989	55,894

Table 7 does not present the true picture of the energy which goes to the ultimate consumer because the energy producing industries themselves are large consumers of energy. It is desirable to trace transactions of energy exchange, and account for the quantity of energy consumed in all categories by each energy producing industry, the remainder of such production being that which actually reaches consumers outside of the energy production circle. Table 8 presents such breakdown.

It is evident from Table 8 that energy producing industries themselves consume an appreciable amount of their production. Such consumption does not reflect the additional large allowance which is consumed in energy transport by railroads, tankers, etc.

By detailed procedure of tracing and evaluating energy consumption by groups or commodities, it is possible to apportion the energy distribution as well as to estimate future energy consumption in terms of production trends. The

TABLE 7—PRIMARY ENERGY SUPPLY—PRODUCTION AND IMPORT DATA FOR 1947[5]

	U. S. BUREAU OF MINES CIRCULAR 7582 VALUES 10^{15} BTU			
SOURCE	PRODUCED	IMPORTED	TOTAL	PERCENTAGE TOTAL
Bituminous Coal	16,532	8	16,540	44
Anthracite Coal	1,556	—	1,556	4
Crude Oil	11,142	585	11,727	31
Natural Gas	6,074	—	6,074	16
Natural Gasoline	NP	—	—	—
Coke Oven	NP	3	3	—
Manufactured Gas	NP	—	—	—
Refineries	NP	371	371	1
Electric Power				
Hydro-generated	1,417	22	1,439	4
Fuel-generated	NP	—	—	—
Total	36,721	989	37,710	100

TABLE 8—ENERGY DISTRIBUTION FOR 1947[5]

U. S. BUREAU OF MINES CIRCULAR 7582
VALUES 10^{12} BTU

SOURCE OF USE	ENERGY	PERCENT OF PRIMARY SUPPLY
Use and Loss—Energy Producing Industries	5,530	15
Use and Loss—Non-Energy Producing Economy:		
Bituminous Coal	11,250	30
Anthracite Coal	1,363	4
Crude Oil (Exports Only)	278	1
Natural Gas	2,762	7
Coke Oven	2,056	5
Manufactured Gas	371	1
Refineries	10,473	28
Electric Power	3,627	9
Total	37,710	100

Bureau of Mines studies provide a wealth of detail and only one specific case of a free-enterprise, free-choice economy is presented in Table 9 to compare 1947 energy figures with the projected figures of 1965.

TABLE 9—ENERGY CONSUMPTION BY CONSUMERS FOR 1947 AND 1965 ESTIMATE[5]

U. S. BUREAU OF MINES CIRCULAR 7582
VALUES 10^{12} BTU

CONSUMER	1947		1965		% INCREASE
	ENERGY	% TOTAL	ENERGY	% TOTAL	
Energy Producing Industries	5,531	14.7	8,175	16.8	48
Manufacturing (1939 Census Groups)					
Iron and Steel	3,045	8.1	4,428	9.1	45
Chemicals	1,596	4.2	3,180	6.6	99
Textile	396	1.1	479	1.0	21
Stone, Clay, Glass	899	2.4	1,281	2.6	42
Food and Allied	700	1.9	891	1.8	27
Paper and Allied	654	1.7	865	1.8	32
Nonferrous Metals	481	1.3	867	1.8	80
Other Mfg.	1,220	3.2	1,977	4.1	62
Total	8,990	23.9	13,968	28.8	55
Railroads	3,736	9.9	2,063	4.3	−45
Bunker	803	2.1	935	1.9	16
Household, Commercial and Other Domestic Fuel and Power					
Heat	8,182	21.8	10,846	22.3	33
Power	1,975	5.3	3,559	7.3	80
Motion	4,391	11.7	6,021	12.4	37
Total	14,548	38.7	20,426	42.1	40
Domestic Chemical, Lubes, etc.	969	2.6	1,372	2.8	42
Exports	3,025	8.0	1,600	3.3	−47
Total	37,602	100.0	48,539	100.0	29

There are many conclusions which could be drawn from the available data, but the most significant appears to be the probable increases in an 18-year period of a 30 percent demand upon our energy resources. Projected, these figures would infer an energy consumption doubling in each 50 years, which, in turn, infers a U. S. energy consumption in the year 2000, equal to the current consumption of the entire world, or, say, 25 trillion horsepower-hours per year. It would also appear that in terms of such large demand, real attention should

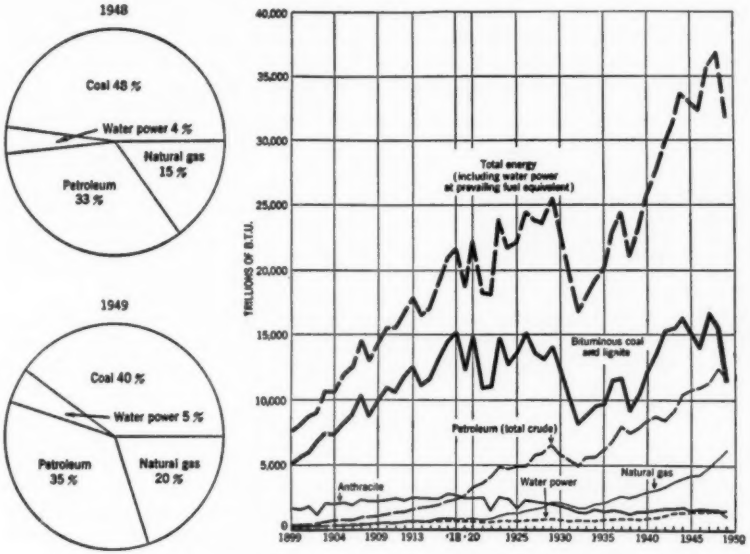


FIG. 3. ENERGY PRODUCTION IN THE UNITED STATES

be directed toward substitution of the continuous sources of energy for the discontinuous, if for no other reason than national conservation.

ENERGY FROM VEGETATION

It is currently estimated that wood is consumed as fuel to the extent of about a trillion horsepower-hours per year and that the world power consumption could be provided by about 3 million square miles of forest, or about 5 percent of the land area of the earth. Actually, the earth produces about 30 billion tons of vegetation which would be equivalent to some two and one-half times our world energy requirement. The direct use of vegetable matter, however, is difficult from an economic standpoint, primarily because of the labor involved in harvesting and the necessity of storage. As our economy depends more and more upon regulated cropping of forest resources, and dependable supplies of

waste wood at fixed locations develop, we will no doubt make better use of wood as a fuel source.

Indirectly, the production of alcohol or other hydrocarbon materials from vegetable matter affords a field of exploitation which gives energy in transportable form as liquid fuel. Subject to the priority of food crops, vegetable matter could, and undoubtedly will, be grown and converted to liquid fuel. One ton of dry wood yields about 84 gal of alcohol. Wood, as well as coal, can be converted to gas for conventional synthesis to gasoline and other hydrocarbons. From an economic standpoint, it would be more reasonable to convert wood into gasoline upon a strictly thermal basis. Much depends upon uses of alcohol other than fuel, however, and, in addition, upon the use of lignin, waxes, and other

TABLE 10—CONTINUOUS SOURCES OF ENERGY[1]

SOURCE	TRILLION HORSEPOWER-HOURS PER YEAR PROBABLE 100 YEARS HENCE
Vegetation	10.0
Solar Power	20.0
Solar Heating	4.5
Hydroelectric	6.0
Wind	Trace
Heat Pumps	0.2
Tropical Waters	0.7
Earth Heat	0.5
Tides	0.05
	41.9

by-products of the wood sugar fermentation process. Both conversions of wood will undoubtedly, play a distinct part in the future.

In connection with generalized usage of vegetable substances, photosynthesis research holds possibilities for greatly accelerated rates of growth through the catalytic reaction of carbon dioxide and water. It is highly probable that a constantly renewable source of energy of some magnitude may result from such researches.

SOLAR ENERGY

Essentially, all energy, continuous and discontinuous, stems from an atomic source, and but for a very few exceptions, is transmitted and captured through solar radiation. The largest energy source is the sun from which we receive only a small fraction of the radiation. Even this small fraction is 10^5 times our total energy requirement. About half of the intercepted radiant energy is reflected back into space by our atmosphere and another third of the remainder by the earth's surface. Part of the unreflected energy is fixed by plants and utilized in evaporation of water, and the remainder is re-radiated at night. Effective utilization of solar energy is essentially a problem of preventing such re-radiation by trapping the solar radiation in some usable form.

It is quite possible, at very low efficiencies, to convert solar radiation to electrical energy via thermoelectric means and new alloys have recently raised

efficiencies to as high as 5 percent. Likewise, photoelectric conversion holds definite possibilities. Primarily, however, solar energy potential is raised by optical systems and several such systems have been projected. High temperature production as well as low pressure power systems are a distinct possibility, and some estimates have been prepared as to costs and the land areas involved. Night storage of solar energy and storage to meet peak demands are difficult but surmountable problems. Direct use of solar heat by solar collectors built directly into dwelling structures has been the subject of extensive experimentation and is entirely feasible as a heating system, particularly since new glasses have been developed to permit higher efficiencies of energy transmission. It is considered that, south of latitude 40, houses can now be designed and built to require no fuel for domestic heat and hot water and that the overall cost over the life of the structure is comparable to that of the conventional dwelling with its yearly fuel bill.

WIND POWER

The experiments at Grandpa's Knob, Vermont, and other less comprehensive experimentation have indicated possibilities of air movement as a source of power. Variable wind velocities pose difficulties, with winds below 20 mph being of little practical use, and winds over 30 mph presenting serious structural difficulties. Storage of power for peak demand is a difficulty, and any installation would have to be integrated with an interconnected power grid to be practical. Investment costs have been figured at about $68 per kilowatt. Although not of probable great importance, wind power experimentation still continues.

EARTH HEAT

The earth radiates more heat than it receives from outer space. This heat comes from the hot interior of the earth (variously estimated at from 2,000 to 20,000 deg C) and is about 250 trillion horsepower hours per year. No practical plan has so far been developed to use earth heat, except in those few locations where vulcanism brings high temperatures close to the surface and where volcanic steam is present. The largest installation is in Italy, where about 170 million horsepower-hours per year are generated. This plant has recently undergone extensive expansion. The engineers of the future will perhaps conceive more plausible ideas for the utilization of earth heat than they have in the past, for a large storehouse of energy is present if it can be tapped in a practical and economic manner.

TIDAL POWER

Tidal power is one of the non-atomic sources of energy, and, if totally harnessed, might provide about half of the total energy requirement of the world. There are, however, only a few points on continental shores which appear to justify installations. Of the dozen or so probable locations, a few are very interesting and the potential appears to justify future plants of some appreciable magnitude at these sites. For example, the Severn River in the British Isles, Passamaquoddy, and the Bay of Fundy, Canada, are distinct probabilities. These three projects alone might produce about 24 million horsepower-hours per year. The

absolute practical potential of tidal power as now conceived could not exceed 0.5 percent of our world requirement.

TROPICAL WATERS

Because of the difference in temperature between the surface and depths of tropical waters, a power potential exists which has been the subject of extensive study. A plant of 7,000 kw capacity is being constructed on the coast of French West Africa, and only recently an expansion of this operation has been proposed. It appears reasonable that supplemental power installations using water temperature differentials will be constructed, but the total magnitude will not be large due to restricted locations. The total generation, however, may probably exceed that from tidal power. It is quite possible that total power from tropical waters could eventually equal that of total hydro at the present time.

HEAT PUMP

Although it is designated as an energy source, the heat pump or *reversed cycle heating* is a mechanism by which the difference in temperature between atmosphere and earth is used for deriving power usually as heat energy. Since the heat pump is a mechanism and not a source discussion of it should be limited to a single simple statement. Extensive studies have greatly improved techniques in use of the heat pump. Today both the equipment and ability are available to take advantage of situations in which *invested* energy can be made to yield an increased increment where earth or water heat sources are adequate and the economies are favorable. In the future heat pump installations may yield an increment of energy, perhaps as high as 0.2 trillion horsepower-hours per year.

CONCLUSIONS

The foregoing by necessity is a very brief coverage of the subject of energy sources. Combined with the detail of many studies, however, it would appear that certain conclusions are warranted, specifically:

1. With the exception of hydro-electric power, our energy comes from discontinuous sources, the fuels of which are finite in amount.

2. That our economy is expanding in proportion to our energy use, and we can expect greater demands upon our energy sources.

3. That, while there are ample and available sources of energy to provide for our immediate requirements, and that such sources can be expanded to an appreciable degree, they are nevertheless, finite.

4. In the interest of future generations as well as in the interest of true conservation, we should hoard our fossil fuels wherever possible, particularly petroleum.

5. To conserve our fossil fuels infers substitutions of one discontinuous source for another and continuous sources for the discontinuous ones.

6. That there are distinct possibilities of greater development and use of continuous sources of energy which Ayres[1] has estimated as probable within 100 years as shown in Table 10.

7. That energy sources, limitations, and locations may well influence our living patterns through change in architectural design, industrial locations, comparative price of fuels, and the development of more integrated energy systems.

8. In addition to development of new sources of energy, we have much to accomplish by way of improvement in use of currently developed sources through higher efficiencies of utilization, transport, and conversion.

9. That periodic inventory of our energy resources is just as important as for other natural resources.

10. That we should be willing to pay a premium now in terms of research and development to effect conservation of our *least adequate in amount, but most useful in form* sources of energy. Such research as shale oil recovery, liquid fuels from coal, utilization of vegetable wastes, etc., should continue to receive adequate public support.

11. That we should direct conservation effort preferentially to areas of largest consumption such as space heating and provision of process heat.

12. That, in the public interest, we should eliminate wasteful processes where conservation could be effected by substitution even if such substitution results in a temporary economic disadvantage.

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Variables Affecting Solar Incidence, by L. B. Anderson.

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In Memoriam 1951

Name	Joined
William W. Allen, Jacksonville, Fla.	1936
William H. Ballman, Baltimore, Md.	1937
Louis N. Beers, Tucson, Ariz.	1942
John G. Blanchard, Cincinnati, Ohio	1950
Isaac M. Bortman, Washington, D. C.	1946
Robert E. Breisch, Santa Monica, Calif.	1951
James H. Carson, Miami, Fla.	1946
Durno Chambers, New York, N. Y.	1948
Joseph R. Darnell, Milwaukee, Wis.	1949
Thomas M. Dugan *(Life Member),* McKeesport, Pa.	1920
Arthur B. Fels *(Life Member),* Portland, Me.	1919
Nelson B. Fenton, Jackson, Mich.	1949
Edward W. Gibbs *(Life Member),* Providence, R. I.	1919
Alfred H. Griewisch, Milwaukee, Wis.	1938
Elliott D. Harrington, Schenectady, N. Y.	1930
Charles A. Hawk, Jr., Pittsburgh, Pa.	1949
Joseph G. Hayes *(Life Member),* Indianapolis, Ind.	1908
W. Stanley Lott, Walla Walla, Wash.	1949
Charles E. Lucke *(Life Member),* New York, N. Y.	1924
Dayle G. Malone, Chicago, Ill.	1925
George W. Martin *(Life Member),* Ridgewood, N. J.	1911
Walter L. Mayne, Cincinnati, Ohio	1938
Claude E. McCoy, Little Rock, Ark.	1936
James F. McIntire *(Life Member* and *Presidential Member),* Detroit, Mich.	1914
Leonard A. Michael, Denver, Colo.	1945
Arthur S. Morgan, Toronto, Ont., Canada	1937

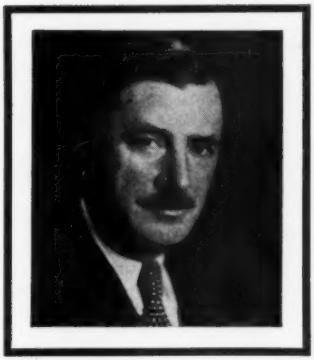
In Memoriam 1951

(*continued*)

NAME	JOINED
Edward T. Murphy *(Life Member)*, San Francisco, Calif.	1915
Haydn Myer, Birmingham, Ala.	1940
Walter W. Nobbs *(Life Member)*, London, England	1919
Carl K. Otis, Buffalo, N. Y.	1948
Louis H. Pogalies, Cleveland, Ohio	1931
Howard W. Pound, Louisville, Ky.	1941
Harry L. Repp, Cleveland, Ohio	1940
Edmund J. Ritchie, New York, N. Y.	1923
Henry L. Roberts *(Life Member)*, Philadelphia, Pa.	1916
Fred H. Schernbeck, Minneapolis, Minn.	1930
August H. Schroth *(Life Member)*, San Gabriel, Calif.	1911
Charles L. Templin, Atlanta, Ga.	1921
Arthur W. Thompson, St. John, N. B., Canada	1949
Roy D. Tyler *(Life Member)*, Bellows Falls, Vt.	1928
Walter E. Voisinet, Buffalo, N. Y.	1930
John S. Webb, Boston, Mass.	1945
Robert P. Whitmer, Bloomington, Ill.	1935
John A. Wilson, Columbus, Ohio	1948

James F. McIntire

1882-1951



JAMES F. McINTIRE, former president of the United States Radiator Corp., and a past president and Life Member of the Society, died June 18, in St. Petersburg, Fla. He was one of the leading figures in the industry and was extremely active in A.S.H.V.E. work both nationally and in the Michigan Chapter.

Born on July 13, 1882 in Farmer City, Ill., Mr. McIntire was graduated with a B.S. degree from the University of Illinois in 1907. He began his business career with J. S. Farrell and Co., Indianapolis, Ind., as superintendent of the plumbing and heating departments. He then was engaged by the United States Heater Co., as salesman, engineer, and draftsman, remaining with that company until June, 1910, when he joined the United States Radiator Co. His first assignment was on boiler design and test engineering and marked the start of an outstanding career, which included service as chief engineer from 1918 to 1924, as vice president for many years, and which was climaxed in 1945 with his election as president. Mr. McIntire supervised the Corporation's research activities and also served as the firm's representative to I=B=R, assisting in the formulation of numerous codes and regulations.

He retired from United States Radiator in 1948, but continued to serve in an advisory capacity. Mr. McIntire was recognized as an authority on cast-iron boiler design, and was one of the pioneers in developing new radiator types.

Mr. McIntire joined the Society in 1914 and served as second vice president in 1937, first vice president in 1938, and president in 1939. In addition, he was

a member of the Council for nine years and served on many committees. Among these were the following:

1918-24 Code for Testing Low Pressure Heating Boilers
To Confer with *A.S.M.E.* Boiler Code Committee on Heating Boilers
Code of Minimum Requirements for Heating and Ventilation of Buildings
Sub-Committee VII—Heating Boiler Capacity (chairman—1923)
Code of Heating and Ventilating
Sub-Committee VI—Heating Boiler Capacity (chairman—1924)

1925-31 Guide Publication
Code for Testing Low Pressure Heating Boilers
To Confer with *A.S.M.E.* Boiler Code Committee on Heating Boilers
Code of Heating and Ventilating
Sub-Committee VI—Heating Boiler Capacity (chairman)
Finance
TAC on Radiation
Executive
Rating and Selection of Low-Pressure Heating Boilers
Sub-Committee on Load Determination and on Chimney Selection
Advisory Committee on International Heating and Ventilating Exposition
Committee on Research
Executive Committee of Committee on Research
TAC on Radiation—Direct and Indirect
Committee on Definitions
Nominating Committee—Michigan Chapter representative

1932-38 Committee on Research
Executive Committee of Committee on Research
TAC on Gas Heating Equipment
Chapter Relations
Code for Testing and Rating Concealed Gravity Type Radiation
Meetings
TAC on Direct and Indirect Radiation with Gravity Air Circulation
Nomenclature
Finance (chairman 1937 and 1938)
A.S.H.V.E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers
Radiation with Gravity Air Circulation IP-9
F. Paul Anderson Award Committee (chairman——1938)

1939-43 To Study Method of Selecting Society Officers and Council Members
TAC on Radiation with Gravity Air Circulation
Executive (chairman—1940)
F. Paul Anderson Award

The Council expressed the sorrow of the Society and paid tribute to Mr. McIntire's memory in the following resolution:

In memory of James F. McIntire, who served THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS as President in 1939, we, the members of Council, record with sorrow his untimely death June 18, 1951 at St. Petersburg, Florida.

He joined the Society in 1914, and served it long and faithfully as an officer, as the Chairman or member of many of its committees.

We pay tribute to him for his leadership, his farsighted and wise counsel which was important in the growth and progress of the Society.

As an engineer, executive and loyal member devoted to the advancement of his profession we salute his memory.

And, we offer our sincere and heartfelt sympathy to his wife and family whose grief we share.

For nearly four decades Mr. McIntire was an active and interested member who participated in many of the Society's activities that are counted as milestones in its progress. He was a vital force in its research and code work and served ably in both administrative and financial capacities.

Services were held for him in St. Petersburg and interment was in that city. Mrs. McIntire survives and the sincere sympathy of the Society was extended by the Officers and Council.

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